Runaway Truck Arrester System Study

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ABSTRACT

Runaway Truck Arrester System Study

This study, the first phase of an exploratory development study, investigated the technical feasibility through analysis of five alternative runaway truck arresting methods and possible mechanizations. The study objectives were met and the passive arrester, the gravel bed system, was selected as a potential technically sound system and recommended for refinement through testing and preliminary design in a follow-on study phase. Two other active systems, chain and engine brake arresters, were also feasible but suffered from performance limitations and significantly higher system costs compared with the passive system so were not promoted for further development at this time.

RUNAWAY TRUCK
ARRESTER SYSTEM STUDY

CONTENTS

SECTION	TITLE	PAGE
	Abstract	i
	Contents	ii
	List of Figures	iii
	List of Tables	vi
1.0	Introduction	1
2.0	Chain Arrester System	1
3.0	Inertia Wheel Arrester System	6
4.0	Arrester Engine Brake System	11
5.0	Hydraulic Arrester System	21
6.0	Passive Arrester System	25
7.0	Barrier/Vehicle Catchment	31
8.0	Discussion and Recommendations	32
	Appendix	35

LIST OF FIGURES

FIGURE NUMBER	TITLE	PAGE
2-1	Chain Arrester System Installation	1
2-2	Chain Arrester System Operation	2
2-3	Chain Arrester System Setting	2
2-4	Chain Arrester System Performance - Vo = 145 Km/h, GVW = 65,000 Kg	3
2-5	Chain Arrester System Performance - Vo = 145 Km/h, GVW = 7,000 Kg	3
2-6	Chain Arrester System Performance - Vo = 145 Km/h, GVW = 110,000 Kg	4
2-7	Chain Arrester System Performance - Vo = 160 Km/h, GVW = 65,000 Kg	4
2-8	Chain Arrester System Deceleration Capability - Vo = 145 Km/h	5
2-9	Chain Arrester System Stopping Distance - Vo = 145 Km/h	5
3-1	Inertia Wheel Arrester System Installation	6
3-2	Inertia Wheel Arrester System	7
3-3	Inertia Wheel Arrester System Operation	7
3-4	Inertia Wheel Arrester System Setting	7
3-5	Wheel Weight	8
3-6	Wheel Rotation Speed	8
3-7	Maximum Stress Level	9
3-8	Inertia Wheel Arrester System with Separate Tape Reel and Fluid Coupling	10
3-9	Inertia Wheel Arrester System with Tape Reel, Fluid Coupling and Brake	10
4-1	Arrester Engine Brake System	11
4-2	Arrester Engine Brake System Operation	11

LIST OF FIGURES (Continued)

FIGURE NUMBER	TITLE	PAGE
4-3	Arrester Engine	13
4-4	Single Arrester Engine Installation	14
4-5	Dual Arrester Engine Installation	15
4-6	Arrester Engine Operation	16
4-7	Brake Arrester Engine Hydraulic System - Static Pressure Mode	17
4-8	Brake Arrester Engine Hydraulic System - Main Hydraulic Braking Mode	17
4-9	Brake Arrester Engine Hydraulic System - Rewind Mode	18
4-10	Single Arrester Engine Performance	20
5-1	Hydraulic Arrester System Concepts	21
5-2	Elevated Tank Hydraulic Arrester	21
5-3	Tankage PE System	22
5-4	Flow Rate PE System	22
5-5	Velocity Discharge System	23
5-6	Pipe Size and Pressure KE System	23
5-7	Tankage KE System	23
5-8	Tankage PE/KE System	24
5-9	Discharge Pipe Size PE/KE System	24
6-1	Gravel Bed Passive Arrester System Operation	26
6-2	Gravel Bed Passive Arrester System Installation - Section	26
6-3	Gravel Bed Arrester System Performance	28
6-4	Gravel Bed Arrester System Sensitivity	28
6-5	Deceleration Gravel Bed on Uphill Gradient	29
6-6	Stopping Time Gravel Bed on Uphill Gradient	29
6-7	Braking Distance Gravel Bed on Uphill Gradient	29

LIST OF FIGURES (Continued)

FIGURE NUMBER	TITLE	PAGE
6-8	Braking Distance Gravel Bed on Uphill Gradient	29
6-9	Braking Distance Gravel Bed with Chain Arrester	30
8-1	Passive Arrester System Proposed Development Plan - Phase 2 Technical	34

LIST OF TABLES

TABLE NUMBER	TITLE	PAGE
2-1	Chain Characteristics	2
3-1	Component Identification - Inertia Wheel Arrester System	6
3-2	Runaway Truck Kinetic Energy	8
3-3	Allowable Stress Levels	9
4-1	Arrester Engine Braking Systems Data	12
4-2	Runaway Truck Energy	12
4-2a	Component Identification Arrester Engine - Rewind Side	13
4-2b	Component Identification Arrester Engine - Control Side	13
4-3	Component Identification - Single Arrester Engine Installation	14
4-4	Component Identification - Dual Arrester Engine Installation	15
4-5	Periodic Inspection Schedule	19
4-6	Preventive Maintenance Schedule	20
6-1	Gradient Induced Friction Coefficient	28
7-1	Webbing and Cordage	32

1.0 Introduction

On many roads and highways with steep grades, escape lanes are provided for decelerating and stopping downhill travelling runaway trucks. These escape lanes are unsurfaced uphill turnoffs with very steep grades. Braking is provided by the retardation component of gravity and drag of the tires running through the gravel and dirt surface. At some locations, the geography of the hill requires the escape lane to be located on the opposite side of the road to that of the downhill lane. In such a case, a runaway truck driver must risk additional danger of crossing the uphill lane of traffic. The emergency use of an escape lane is, of course, at the discretion of the driver. But, when that option is used, escape lanes are generally effective and result in saving human lives and preventing gross property damage.

Unfortunately, there are potentially dangerous steep grades where escape lanes do not exist today. In some cases, the right-of-way acquisition costs are excessively expensive and in others, the local topography physically prohibits the cutting of a steep escape lane. The purpose of this technical study was to establish the feasibility of an acceptable alternative runaway truck arrester design concept specifically oriented for cases where the standard escape lane is unacceptable.

This was an exploratory development study in which the feasibility of possible alternative runaway truck arresting methods and mechanizations were analyzed. The objective was to select technically sound, safe and cost effective candidate systems, if any, for follow-on refinement and preliminary design. Proposed candidate systems were defined and performance estimates established.

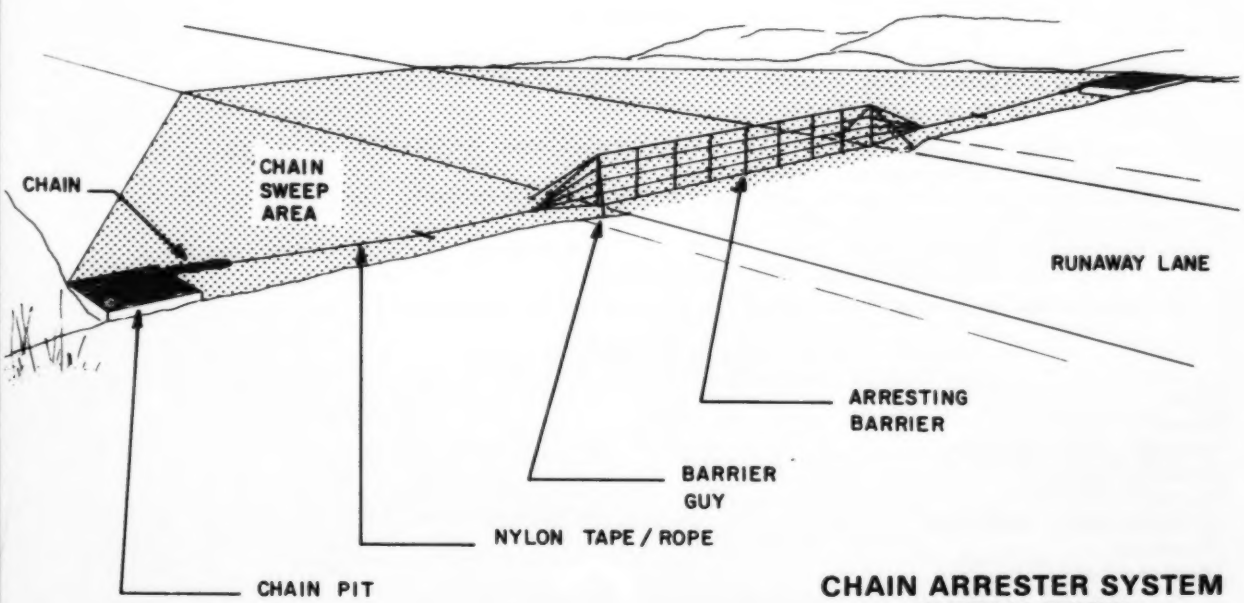
The following runaway truck arrester systems are covered in Sections 2 through 6:

- chain arrester
- inertia wheel arrester
- arrester engine brake
- hydraulic arrester, and
- passive arrester.

The barrier net and vehicle catchment details are presented in Section 7. Conclusions of the comparisons and study recommendations are summarized in Section 8.

2.0 Chain Arrester System

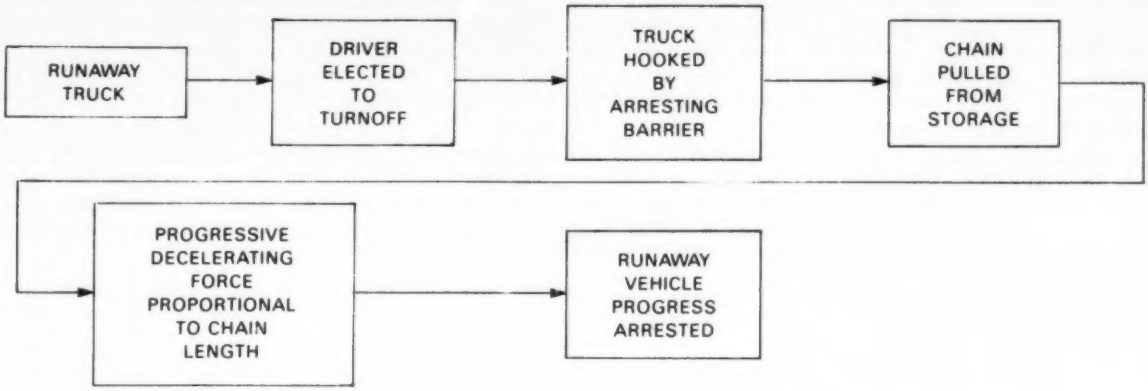
A chain arrester system made up of an arresting barrier with a number of heavy chains attached to each end conceptually is a relatively simple emergency vehicle braking system to mechanize in a roadside turnoff, as shown in Figure 2-1. The arresting barrier is assumed to consist of a resilient nylon line or net. The chain is ship's anchor chain, or equivalent, joined to the barrier with shackles or swivel rings. The chains could be stored in a cavity below the ground surface and stacked for a free pull by the barrier and runaway vehicle. The drag of the heavy chain provides the braking force to arrest the speeding runaway vehicle. After emergency use and vehicle removal, the chains are dragged back, inspected and stacked. The barrier is inspected and set. The tractor used to pull the chain back to storage is the only power required for the system - aside from maintenance, periodic turnoff surface scraping and winter snow removal.



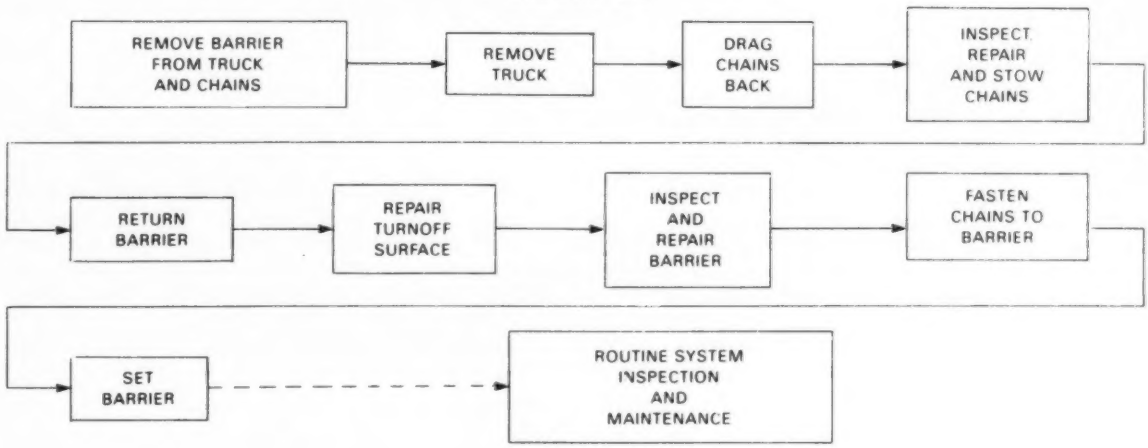
CHAIN ARRESTER SYSTEM INSTALLATION
FIGURE 2-1

2.1 Chain Arrester System Study Assumptions

The operation of the chain arrester system is shown in the flow chart, Figure 2-2, and setting the system after emergency usage in flow chart, Figure 2-3.


RUNAWAY TRUCK
DRIVER ELECTED TO TURNOFF
TRUCK HOOKED BY ARRESTING BARRIER
CHAIN PULLED FROM STORAGE
PROGRESSIVE DECELERATING FORCE PROPORTIONAL TO CHAIN LENGTH
RUNAWAY VEHICLE PROGRESS ARRESTED

CHAIN ARRESTER SYSTEM OPERATION
FIGURE 2-2



CHAIN ARRESTER SYSTEM SETTING
FIGURE 2-3

Assumptions and ground rules for performance analysis of the chain arrester system are:

- truck travels in a straight direction, remains upright and does not jackknife
- the arresting barrier weight is neglected
- frictional forces of the truck tires and the dragging chain are ignored
- truck engine braking is not credited
- reactions are instantaneous
- braking force is provided by the dragged chain weight.

The chain picked for analysis of the system has the following specifications:

TABLE 2-1
CHAIN SPECIFICATIONS

Size		(in)	3
Standard pitch	P	(in)	7-3/4
Average weight per foot		(lb)	86
Outside length	L	(in)	14
Outside width	W	(in)	9-7/8
Average safe working load		(lb)	109,000
Proof test		(lb)	218,000
Approximate braking load		(lb)	436,000

The proof load was tested to U.S. Government and American Bureau of Shipping requirements. The chains are considered to be added evenly to each side of the arresting barrier with a total of 2 (1 per side), 4 (2 per side), 6 (3 per side), 8 (4 per side) and 10 (5 per side).

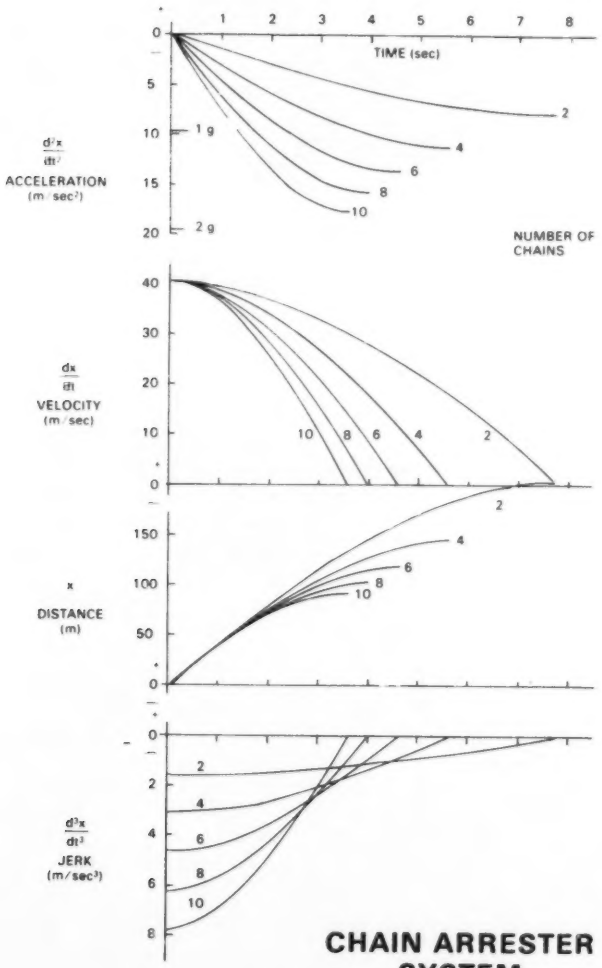
2.2 Chain Arrester System Performance

Results of the chain arrester system performance anaylsis are presented in Figures 2-4 through 2-9.

Figures 2-4 through 2-6 present the performance for the normal design entrance speed of 145 km/h. In each figure is shown the deceleration, the velocity, and the distance travelled histories for the five different number of chains. Also, the third derivative, jerk, history is shown. Although a strict physical interpretation of the peak value of jerk and the nature of the curve is not attempted in this report, the relative values of one case compared to another is observed. For the chain arrester system the deceleration is ramp shaped, satisfying the criterion of paragraph 2 of the Technical Specifications.

From Figure 2-4 for a GVW of 65,000 kg (the maximum normal expected weight) and an entrance velocity of 145 km/h, the peak deceleration range is from 0.8 g (2 chains) to 1.8 g (10 chains). The distance travelled from system engagement is about 90 m (10 chains) to over 200 m (2 chains). The peak value of jerk is almost 5 times higher for the system with 10 chains compared with that of 2 chains and lasts only half as long - a much more severe case.

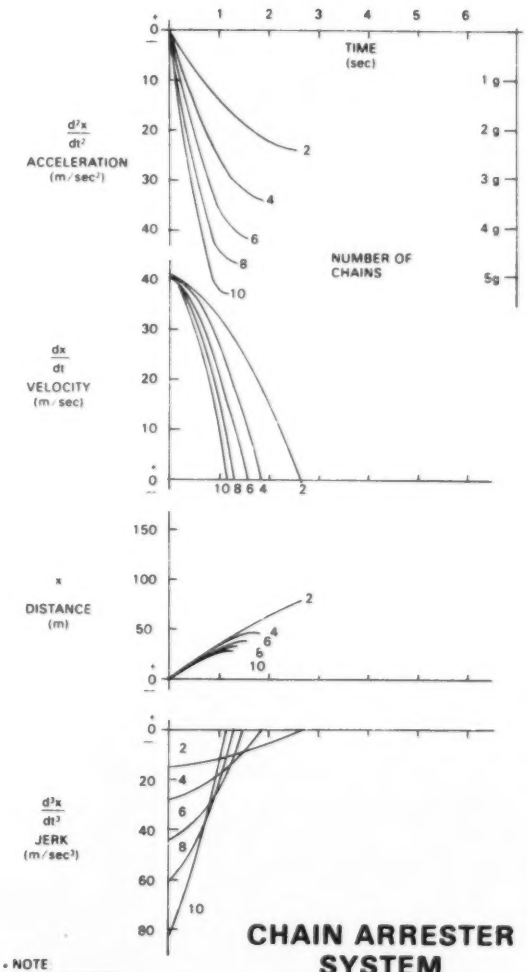
The performance of the chain arrester system shown in Figure 2-5 is for a light truck of 7,000 kg, at the lower end of the expected weight range, at an entrance speed of 145 km/h. It can be seen from this figure that the arrestment is rapid and some severely high levels are reached. The peak deceleration for the 10 chain case is over 5 g and the jerk is an order of magnitude higher than that for the 65,000 kg vehicle. Even the 2 chain case reaches a peak deceleration of 2.5 g, exceeding the study guidelines normal deceleration limit. Because of these high decelerations, the distance travelled is trivial. It is most probable that neither the truck nor its driver, passengers and cargo would survive a stop of such high g unscathed.



CHAIN ARRESTER SYSTEM PERFORMANCE

- Vo = 145 km/h
- GVW = 65,000 kg

FIGURE 2-4



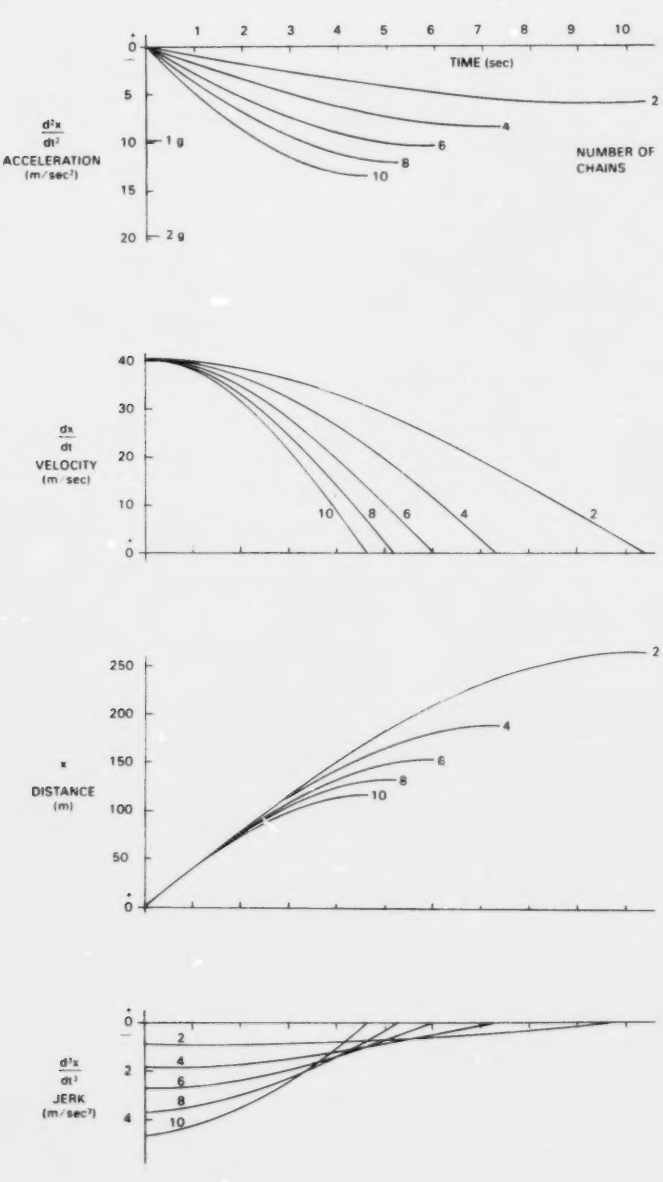
CHAIN ARRESTER SYSTEM PERFORMANCE

- Vo = 145 km/h
- GVW = 7,000 kg

FIGURE 2-5

The same data are presented for the 110,000 kg heavy hauler at an entrance velocity of 145 km/h in Figure 2.6. The deceleration values range from a moderate 0.6 g to 1.4 g. The stopping distances run from 120 m to 265 m and the jerk peak at very comfortable and safe levels, when compared to the 7,000 kg case, of 1 to 5 m/sec^3.

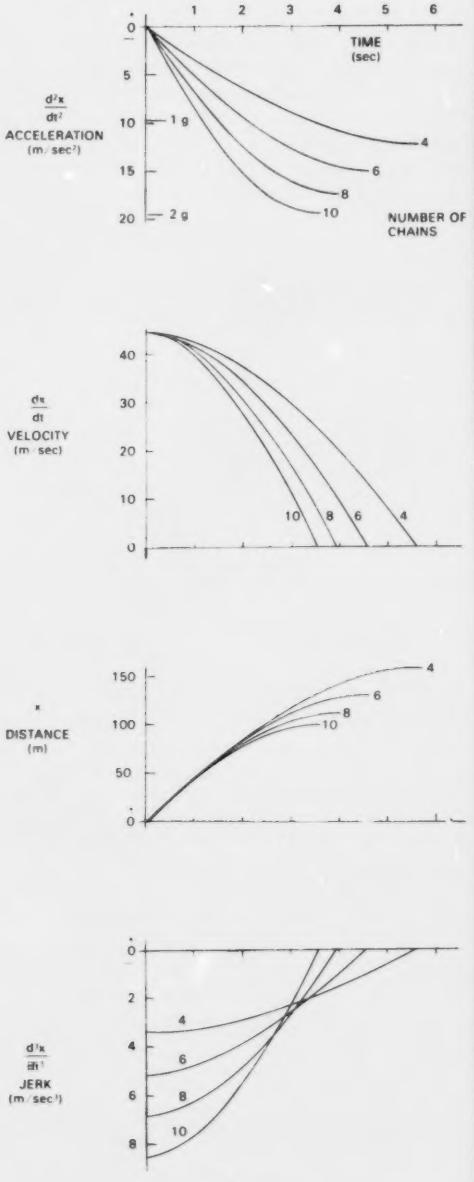
Performance data for the 65,000 kg vehicle at a maximum entry speed of 160 km/h are presented in Figure 2-7. The 10 chain case reaches a peak deceleration of 2 g and a corresponding stopping distance of about 100 meters. The peak value of jerk is almost twice that of the 110,000 kg heavy hauler with an entrance speed of 145 km/h. The 65,000 kg vehicle with an entrance speed of 160 km/h exhibits a 10 per cent increase in the parameters shown relative to those of the vehicle at a normal expected entrance speed of 145 km/h.



CHAIN ARRESTER SYSTEM PERFORMANCE

- Vo = 145 km/h
- GVW = 110,000 kg

FIGURE 2-6



CHAIN ARRESTER SYSTEM PERFORMANCE

- Vo = 160 km/h
- GVW = 65,000 kg

FIGURE 2-7

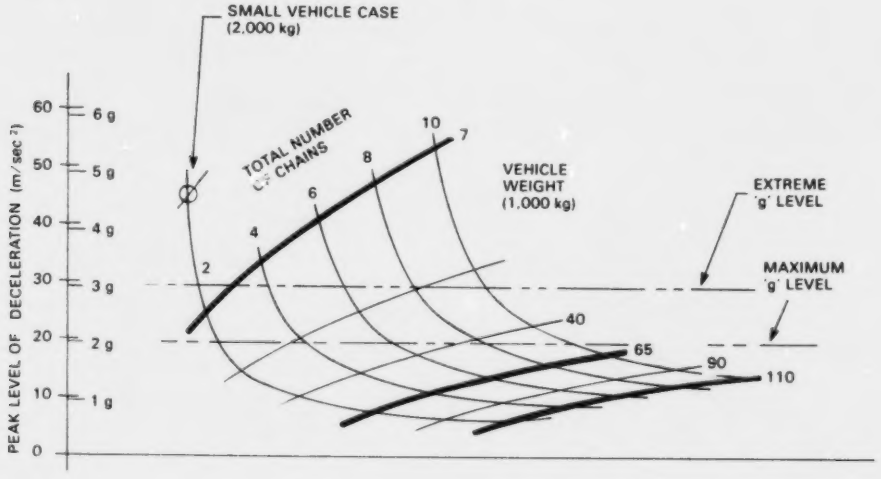
2.3 Chain Arrester System Operational Capability

The deceleration capabilities of a chain arrester system are summarized in Figure 2-8. The heavier the chain weight, the heavier is the vehicle that can be accommodated within the maximum 2 g level. For example, with a 10 chain system the deceleration of any vehicle weighing less than 50,000 kg will exceed the maximum 2 g limit and the extreme 3 g limit will be exceeded by vehicles weighing less than 23,000 kg. For the 2 chain system all vehicles with weights exceeding 10,000 kg will satisfy the 2 g limit but vehicles weighing less than 5,000 kg will have decelerations exceeding 3 g. The deceleration of 2,000 kg small vehicle exceeds both criteria. Its level, at 4.5 g with the 2 chain system, is unacceptable and will have to be catered for in a different manner.

Both the 2 g maximum level and the 3 g extreme level are shown in Figure 2-8. Reading this figure in conjunction with the stopping distance shown in Figure 2-9 a system can be sized for any geographical location. Only the actual braking distance is presented in Figure 2-9. The distance of the lead-in roadway and a far end allowance have to be added to the stopping distance to realize total turnoff length.

A chain 200 m long weighs 56,000 lb. which is below the average safe working load of 109,000 lb. Based on the approximate braking load of 436,000 lb. the factor of safety is 7.8. This chain could be stored in a square pit measuring 2½ m per side.

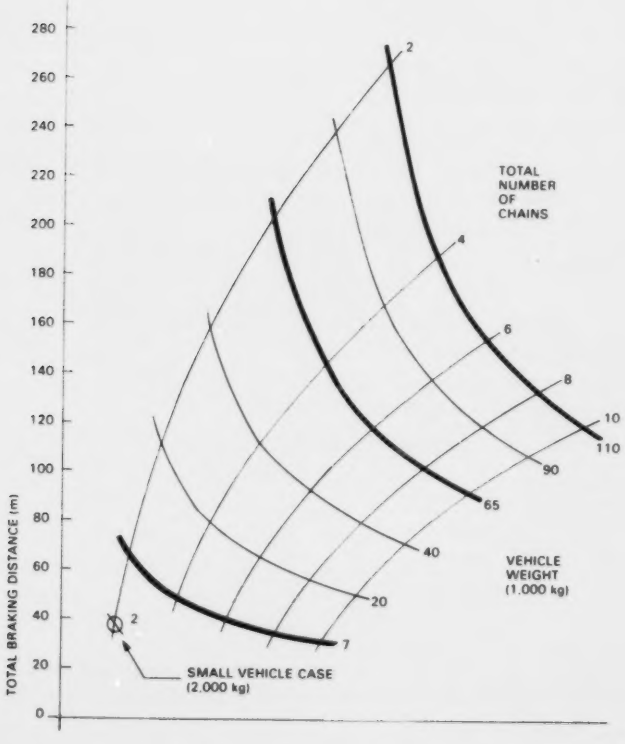
It was concluded that no simple chain arrester system will satisfy all of the criteria laid down in the Runaway Truck Arrester System Study Technical Specifications. Therefore, compromises to this simple system must be made.



CHAIN ARRESTER SYSTEM DECELERATION CAPABILITY

- V_o = 145 km/h

FIGURE 2-8



CHAIN ARRESTER SYSTEM STOPPING DISTANCE

- V_o = 145 km/h
- DISTANCE TO FULL STOP FROM BRAKE APPLICATION

FIGURE 2-9

3.0 Inertia Wheel Arrester System

The change in kinetic energy of a rotating flywheel is given by the equation

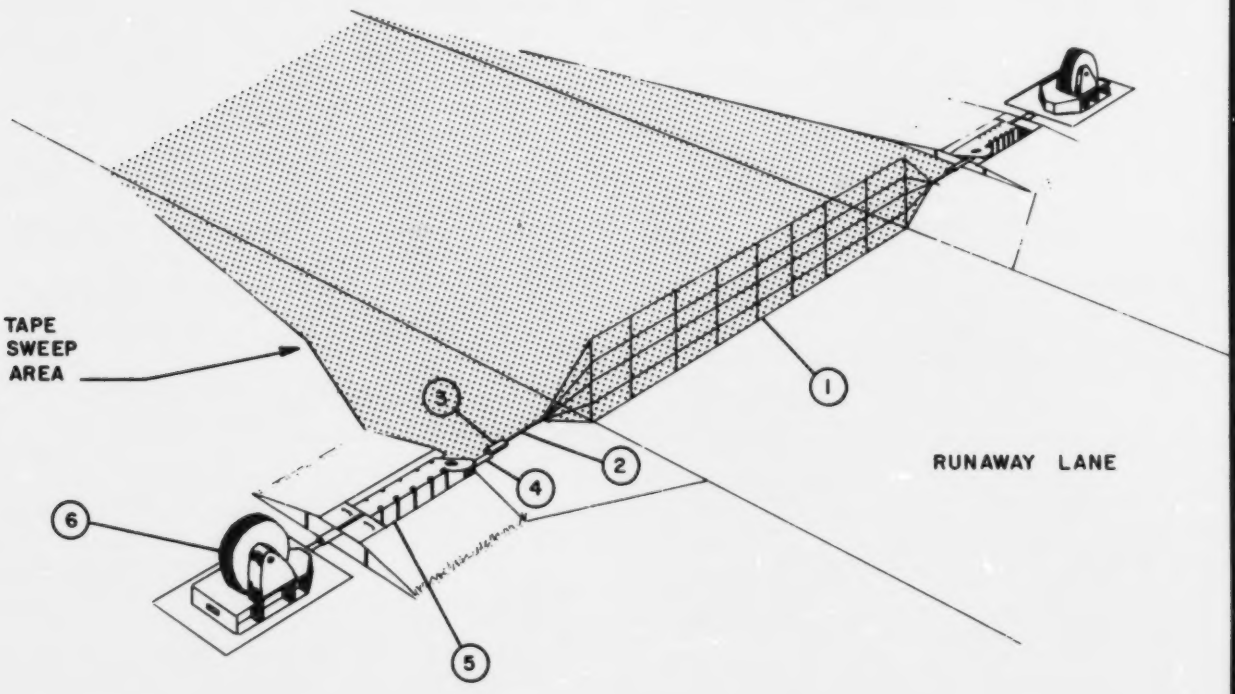
$KE = 1/2\ I\ (W_f^2 - W_i^2)$ where:

I is the mass moment of inertia of the wheel
W_i is the wheel initial rotational speed and
W_f is the final rotation speed.

For a wheel starting from rest, $W_i = 0$, and $KE = 1/2\ I\ W_f^2$. If the moment of inertia and the rotational speed are large, the kinetic energy of a spinning wheel is very high. The purpose of this section is to establish whether or not flywheels could absorb sufficient energy to provide an emergency braking system for a runaway truck.

3.1 System Description

An inertia wheel runaway truck arrester system in its simplist form is assumed to consist of an arresting barrier, like that of the chain arrester system, attached to two flywheels by purchase tapes as shown in Figure 3-1 and Table 3-1.



INERTIA WHEEL
ARRESTER SYSTEM INSTALLATION

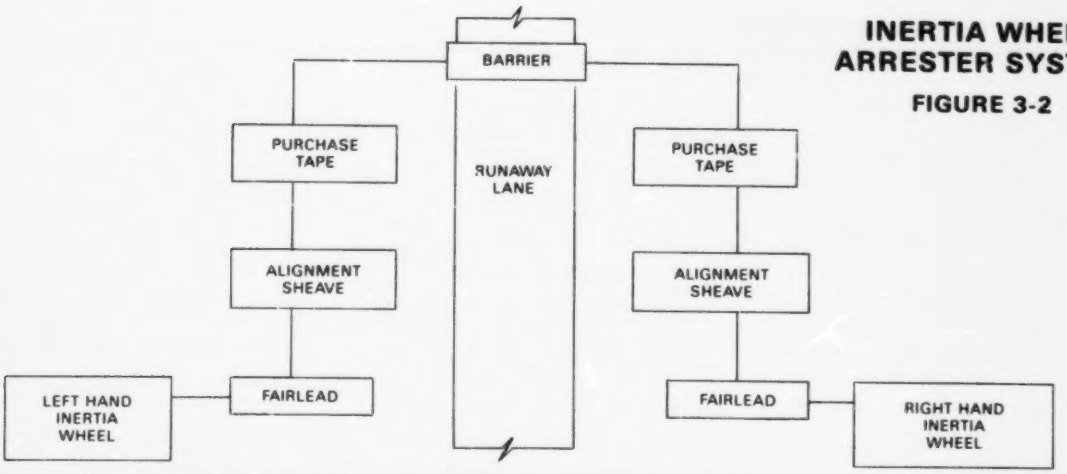
FIGURE 3-1

TABLE 3-1

COMPONENT IDENTIFICATION
INERTIA WHEEL ARRESTER SYSTEM

KEY	COMPONENT
1	ARRESTMENT BARRIER ASSEMBLY
2	BARRIER CABLE
3	TAPE CONNECTOR ASSEMBLY
4	PURCHASE TAPE
5	FAIRLEAD BEAM
6	FLYWHEEL ASSEMBLY

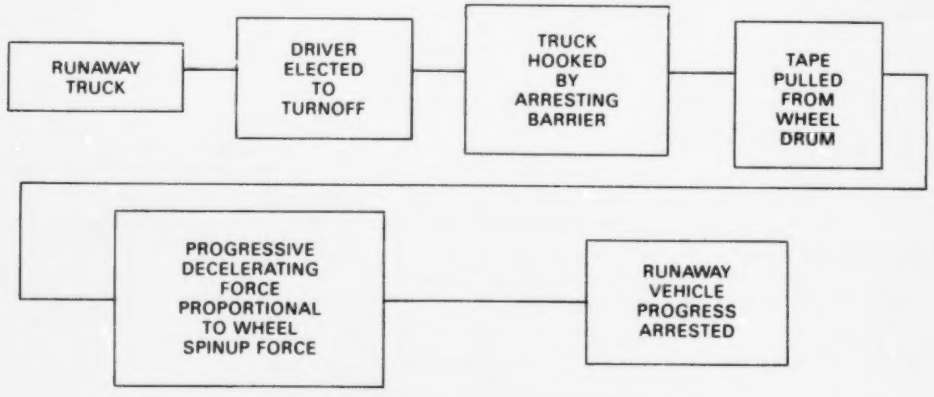
One wheel is mounted on a concrete foundation on each side of a graded runaway lane. A purchase tape is wrapped around the rim of each wheel and aligned with it by a fairlead and alignment sheave. Figure 3-2 is the system block diagram.



INERTIA WHEEL ARRESTER SYSTEM

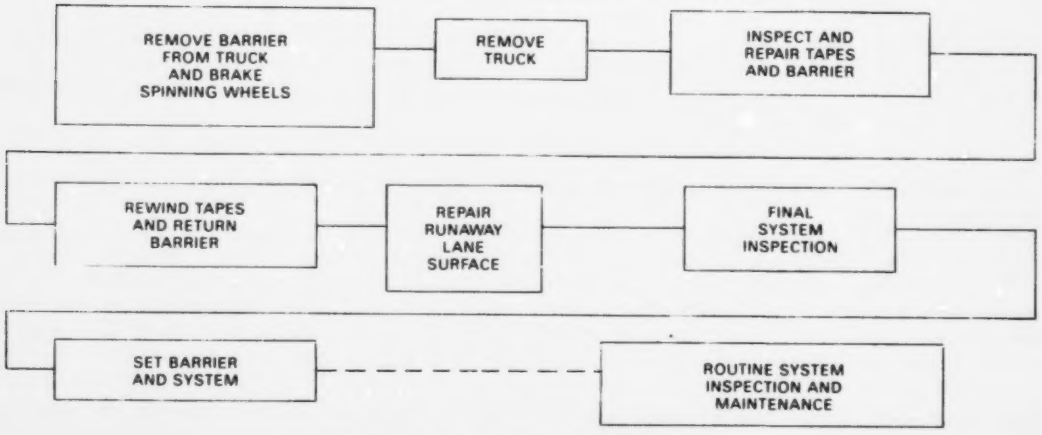
FIGURE 3-2

A runaway truck, upon being caught by the barrier, would rapidly pull the purchase tape from the rim of the wheels and thereby spin them up to speed. It is assumed that the truck would come to rest when the flywheels absorb all of the vehicle's kinetic energy. In this case, the finai velocity of the wheels would be dependent upon the kinetic energy of the runaway truck and the inertia of the arrester system. After truck arrestment, the system would be disengaged, inspected and reset. Runaway lane surface and arrester system maintenance and repair would be conducted. System operation and resetting after emergency use are shown in the flow charts of Figures 3-3 and 3-4.


RUNAWAY TRUCK
DRIVER ELECTED TO TURNOFF
TRUCK HOOKED BY ARRESTING BARRIER
TAPE PULLED FROM WHEEL DRUM
PROGRESSIVE DECELERATING FORCE PROPORTIONAL TO WHEEL SPINUP FORCE
RUNAWAY VEHICLE PROGRESS ARRESTED

INERTIA WHEEL ARRESTER SYSTEM OPERATION

FIGURE 3-3


REMOVE BARRIER FROM TRUCK AND BRAKE SPINNING WHEELS
REMOVE TRUCK
INSPECT AND REPAIR TAPES AND BARRIER
REWIND TAPES AND RETURN BARRIER
REPAIR RUNAWAY LANE SURFACE
FINAL SYSTEM INSPECTION
SET BARRIER AND SYSTEM
ROUTINE SYSTEM INSPECTION AND MAINTENANCE

INERTIA WHEEL ARRESTER SYSTEM SETTING

FIGURE 3-4

3.2 System Performance Evaluation

The kinetic energy for the specified truck gross vehicle weights and entry speeds are summarized in Table 3-2.

TABLE 3-2
RUNAWAY TRUCK KINETIC ENERGY

Vehicle	Weight kg	Kinetic Energy at Entry Speed 10^6 Joules	
		160 km/h (maximum)	145 km/h (normal)
Normal Range	65,000-7,000	64-7	53-6
Light Vehicle	2,000	2	2
Heavy Hauler	110,000	109	89

Consider the wheels to be cast solid disks of steel or malleable iron. Let the wheel thickness be 0.3 metres and the radius range from 0.50 to 2.00 metres. The weight range of the study wheel runs from 2,000 to 30,000 kg and is presented in Figure 3-5. Assuming that the total kinetic energy of the trucks, as shown in Table 3-2, is converted to rotation of the wheels without loss, the final rotation speed ranges from about 50 to 700 radians/second as shown in Figure 3-6. In this figure the rotational speed is shown for the truck normal weight range, the light vehicle and the heavy hauler at the maximum entry speed of 160 km/h and the normal expected entry speed of 145 km/h. The rotational speed due to the energy absorption of the heavier trucks is extremely high. These speeds will impart large internal inertial stresses within the material. The heavy hauler represents the critical design case.

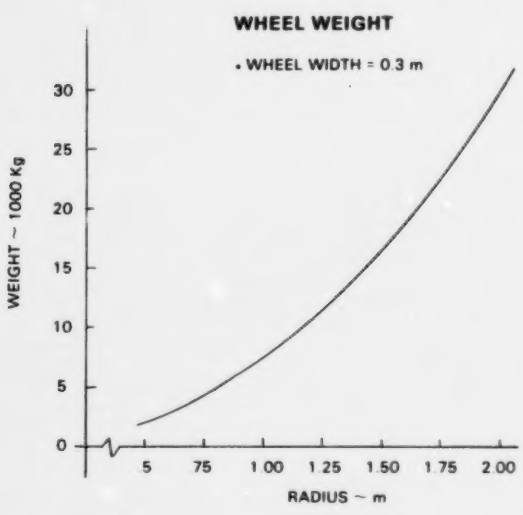


FIGURE 3-5

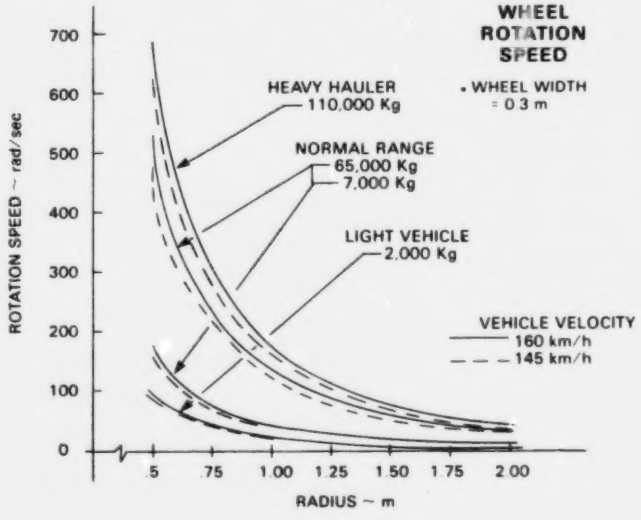


FIGURE 3-6

For elastic conditions, the normal, tangential and radial stresses are equal and, neglecting hub and rim attachment stresses, are

$$\sigma_n = \sigma_t = \sigma_r = \frac{3 + \nu}{8} \rho\omega^2 r^2$$

where ν is Poisson's ratio
ρ is material mass density
ω is rotational speed and
r is wheel radius.

The flywheel stresses, for the heavy hauler case, of the different wheel sizes are shown by the curve labelled "plain wheel" in Figure 3-7. Allowing for a circular cutout at the center of the wheel for an axle at the hub, the tangential stress becomes

$$\sigma_t = \frac{3 + \nu}{4} \rho\omega^2 \left(r^2 + \frac{1 - \nu}{3 + \nu} r_o^2\right)$$

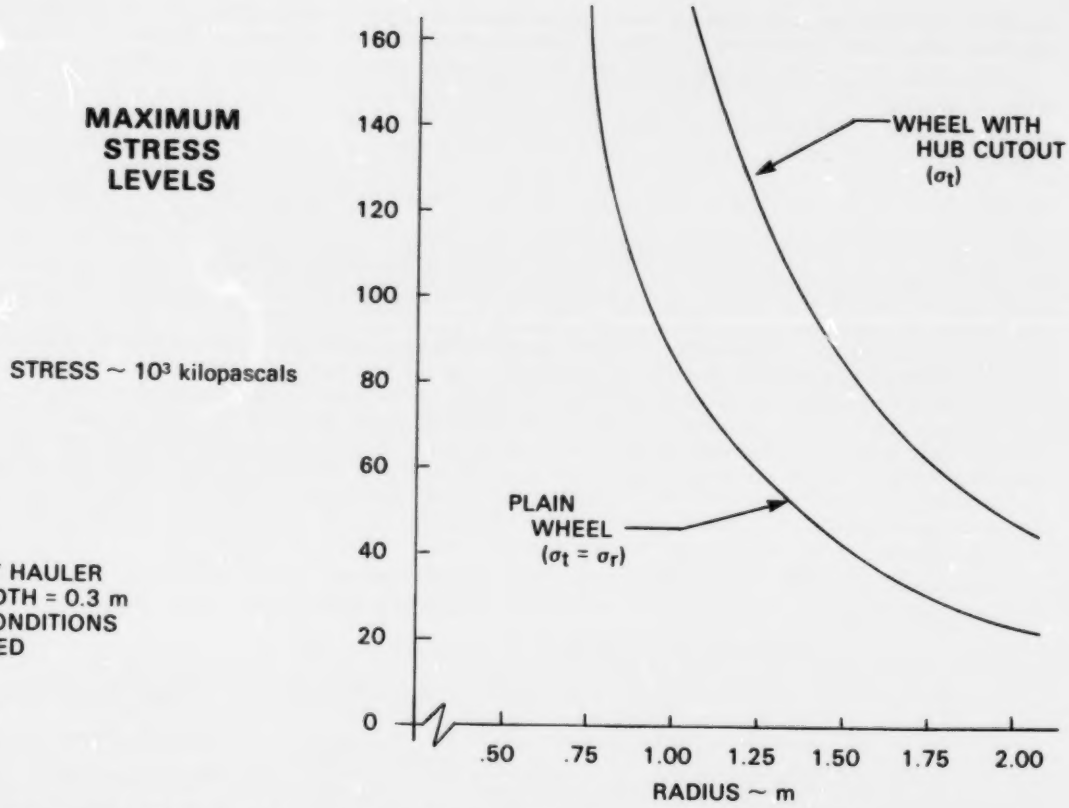


FIGURE 3-7

where r_o is the hole radius. This stress is identified on Figure 3-7 by the label "wheel with hub cutout". The stress levels are higher than those for the plain wheel. In fact, as the radius of the hole approaches zero, the maximum tangential stress approaches a value twice as great as that of the solid wheel; i.e., by having a small circular hole at the center of a solid rotating wheel (for practical purposes) the maximum stress level is doubled. Without proceeding into any further stress details, it is clear that with allowances for stress concentrations and a necessary factor of safety only the large wheel is practical and safe when compared with the allowable stress levels of Table 3-3.

TABLE 3-3
ALLOWABLE STRESS LEVELS

Material	Ultimate Tensile Strength Kilopascals	Elastic limit Kilopascals
Malleable Cast Iron	310,000	103,000
Structural Grade Steel	414 - 496,000	207,000

For energy conservation, impulse must equal momentum; i.e.,

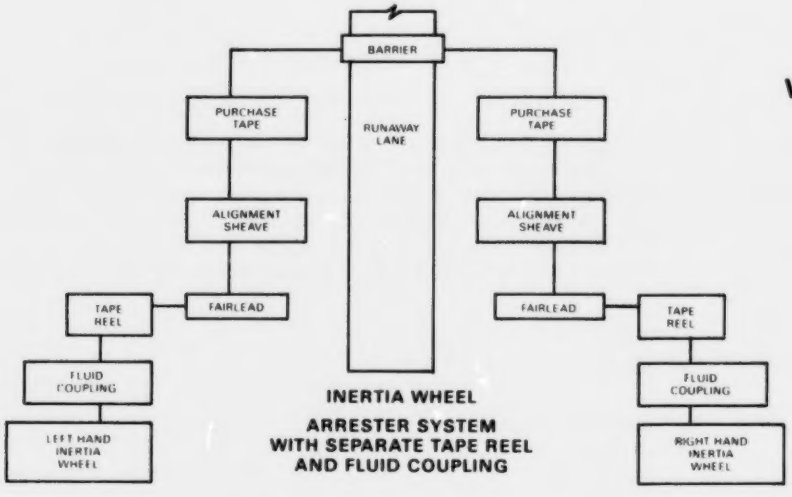
$$\int_{t_o}^{t_l} F\,dt = \int_{v_o}^{v_l} M\,dv.$$

If the tape is considered inelastic the wheel must instantaneously travel at the same tangential velocity as the vehicle is travelling. In this example the acceleration is instantaneous and would impose an infinitely and unacceptably large deceleration upon the vehicle. The decelerating force would greatly exceed the tensile strength capability of the tape. Practically, the wheel cannot accelerate instantaneously to the speed of the vehicle. The difference in the truck velocity and wheel tangential velocity, and the corresponding distances travelled, must be taken up by slip within the arresting system. The purchase tape fabricated from a very elastic material will stretch under load and thereby can provide the apparent slip. Tape made from reinforced woven nylon chords has been developed and found to be satisfactory in aircraft arresting and (parachute) cargo extraction systems. To maintain safe working loads in the purchase tape and provide the stretch for initial wheel acceleration it was found, for the heavy hauler design case, that each tape would have to be several hundred meters long. This, in effect, doubled the land area requirement for the arrester system. It was also found that the large inertia wheels and heavy tapes were too hefty for the light vehicle. With an energy level less than 2% of that of the heavy hauler the inertia wheel system as sized for the heavy hauler would present deceleration loads that

would, like the chain arrester system, inevitably demolish the lighter vehicles. It was therefore concluded that the purchase tape could not provide the total necessary arrester slip for a practical system.

To overcome this performance shortcoming it is assumed that another slip producing element, a fluid coupling, is introduced into the system. The fluid coupling is located between the flywheel and a low inertia tape keeping reel as depicted in the system block diagram Figure 3-8. With the fluid coupling, the low inertia reel could be brought up to speed rapidly in proportion to the vehicle momentum. The flywheel torque and deceleration force could be maintained within allowable values. However, the nature of the deceleration is not right. It is very high initially, peaks and then falls to zero at the time when the reel/flywheel speed is the same as that of the decelerating vehicle; i.e., when the tapes are payed out from the reels at the same rate the vehicle is travelling. Then, there is no longer any decelerating force aside from system friction - the system has reached an equilibrium velocity and in point of fact has broken down. To overcome this design inadequacy and complete the vehicle arrestment another element must be added to the system; a brake, as indicated in Figure 3-9. Assume that the brake is applied to the reel to impede tape pay out and provide the final force to bring the truck to rest. Under this assumption, the whirling flywheel could be disconnected from the reel to keep the brake loads low. This assumption would require another element to be added to the system - a clutch. If the clutch is not incorporated, the brake would have to absorb all of the remaining kinetic energy of the truck (with attached barrier) plus that already absorbed by the inertia flywheel. As a matter of fact, the flywheels will need to be mechanically stopped, otherwise they will rotate for a very long time. For example, if bearing friction is the only retarding force the large wheels will rotate for the best part of a full working day. Obviously a dangerous and unwanted system characteristic. Therefore the brake must be able to absorb all of the system energy and in so doing makes the transfer of the energy to the flywheels and back to the brakes a costly and unnecessary step; i.e., the flywheels are redundant elements of the system.

It is concluded that a simple flywheel arrester system will not perform adequately for a runaway truck emergency braking system. Therefore, it is not considered as a viable system candidate.

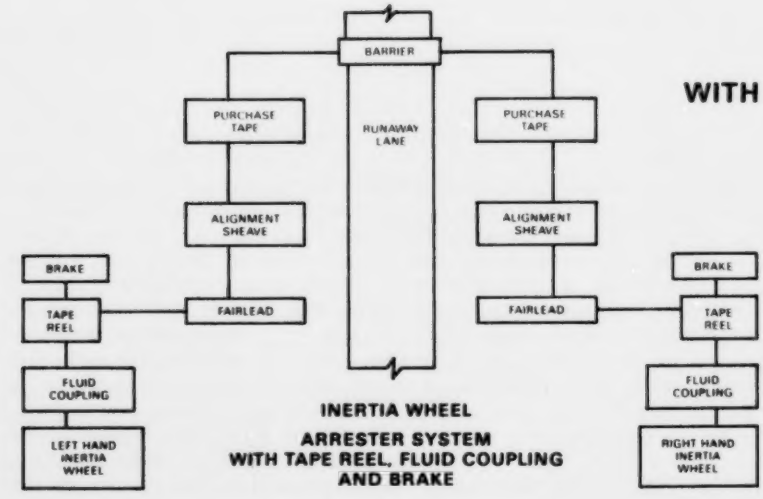


INERTIA WHEEL ARRESTER SYSTEM WITH SEPARATE TAPE REEL AND FLUID COUPLING

INERTIA WHEEL

ARRESTER SYSTEM WITH SEPARATE TAPE REEL AND FLUID COUPLING

FIGURE 3-8



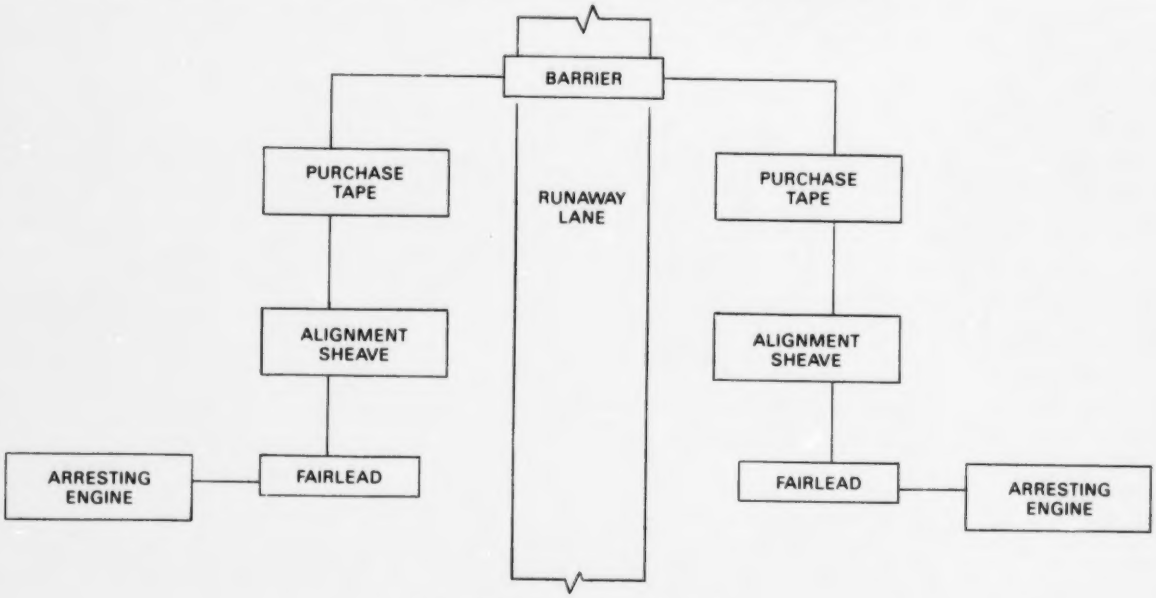
INERTIA WHEEL

ARRESTER SYSTEM WITH TAPE REEL, FLUID COUPLING AND BRAKE

FIGURE 3-9

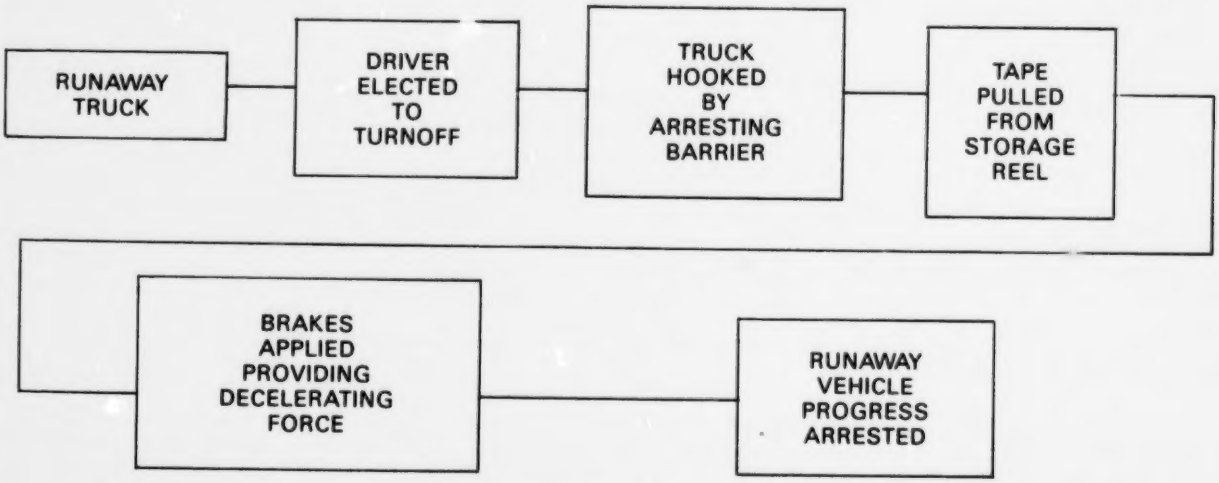
4.0 Arrester Engine Brake System

Elimination of the flywheels from the inertia wheel arrester system results in a basic brake arrester configuration. It can be classed as a rotary friction energy absorber and consists of an arresting engine, fairlead and sheaves, on each side of the runaway lane, connected to the arrestment barrier by purchase tapes. The system is shown in block diagram form in Figure 4-1. Emergency arrestment is accomplished by engagement of a runaway truck with the barrier causing the purchase tape to be reeved from the storage reel, around the entry sheave and through the fairlead. The energy of the arrested runaway truck is absorbed as friction by the rotary brakes during tape pay out. The arrestment process is shown in the flow diagram Figure 4-2. The brake arresting concept described has been adopted as the standard airport runway overrun emergency braking system for military aircraft. As such, it is a highly developed and refined system with much operational use. Several model brake arrester systems were developed and are manufactured by Gulf and Western Industries, Inc., Advanced Development and Engineering Center, Number 4 Horne Drive, Folcroft, Pennsylvania. Data based on the Gulf and Western designs are summarized from operating technical manuals in this section.



ARRESTER ENGINE BRAKE SYSTEM

FIGURE 4-1


RUNAWAY TRUCK
DRIVER ELECTED TO TURNOFF
TRUCK HOOKED BY ARRESTING BARRIER
TAPE PULLED FROM STORAGE REEL
BRAKES APPLIED PROVIDING DECELERATING FORCE
RUNAWAY VEHICLE PROGRESS ARRESTED

ARRESTER ENGINE BRAKE SYSTEM OPERATION

FIGURE 4-2

Basic system description of several candidate Gulf and Western braking systems are presented in Table 4-1. The data description is for production systems configured for aircraft arrestment.

TABLE 4-1
ARRESTER ENGINE BRAKING SYSTEMS DATA

ITEM	DATA			
	152 cm REEL	168 cm REEL	DUAL 168 cm REEL	DUAL 183 cm REEL
Gross Weight (Kg)	8,400	8,400	17,700	17,700
Arresting Engine Dimensions:				
Length (cm)	318	318	318	318
Width (cm)	132	132	132	135
Height (cm)	155	170	170	185
Energy Absorbing Capability (Joules)	88 (10^6)	115 (10^6)	230 (10^6)	407 (10^6)
Vehicle Weight (Kg)	18,144	22,680	45,360	63,500
Maximum Engaging Velocity (km/h)	352	352	334	334
Runout Distance (m)	290	366	366	610
Rewind System	Gasoline Engine	Gasoline Engine	Gasoline Engine	Gasoline Engine
Rewind Time (min)	3	3.5	10	5
Ambient Temperature Range (°C)	-40 to 50	-40 to 50	-40 to 50	-40 to 50
Fuel	Avgas 90 Octane	Avgas 90 Octane	Avgas 90 Octane	Avgas 90 Octane

The significant observation is that the aircraft weights are lower and velocities higher than those of the trucks. The maximum aircraft weight is 58 per cent of the heavy hauler weight and 98 per cent of the weight of a truck at the upper end of normal expected weight range. The maximum expected engaging velocity of the truck is 48 per cent of design value for the aircraft and the normal expected entry speed is only 43 per cent. For the runaway truck design cases, at the 160 km/h maximum expected entry speed, the total kinetic energy is summarized in Table 4-2.

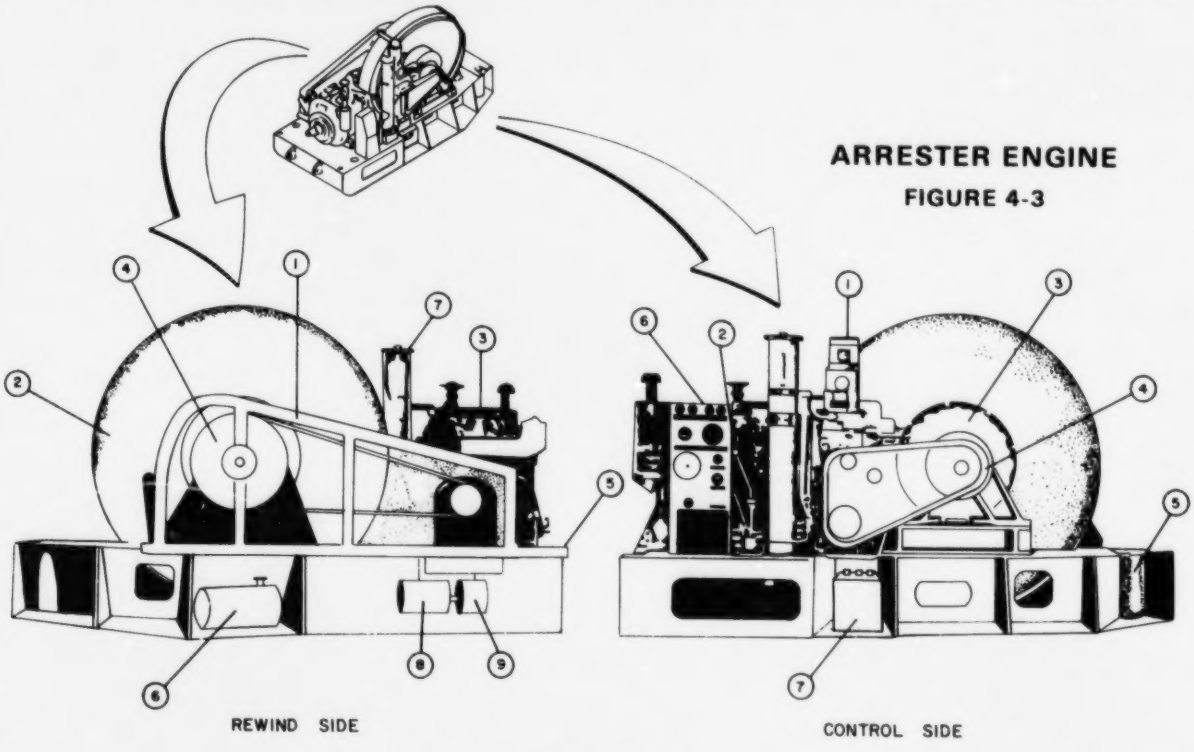
TABLE 4-2
RUNAWAY TRUCK ENERGY

TRUCK WT (Kg)	KINETIC ENERGY (JOULES)
110,000	109 (10^6)
65,000 - 7,000	64 (10^6) - 7 (10^6)
2,000	2 (10^6)

Although the production arrester engines are set up for a different weight and velocity schedule, the energy absorption capability exceeds the energy levels of the runaway trucks. All four arrester engines used as examples are capable of handling trucks up to 65,000 Kg. The three larger units can handle vehicles up to 110,000 Kg. All four units will essentially handle all of the truck weights at the normal 145 km/h entry speed. Consequently, a more detailed review and analysis of arrester engine braking systems is warranted and presented in the following sections.

4.1 Arrester Engine System Description

An introductory description was made in the opening paragraphs of this section. Following is a more detailed description. Reference is made to Figure 4-3 and Tables 4-2a and 4-2b for a typical arrester engine configuration.



ARRESTER ENGINE

FIGURE 4-3

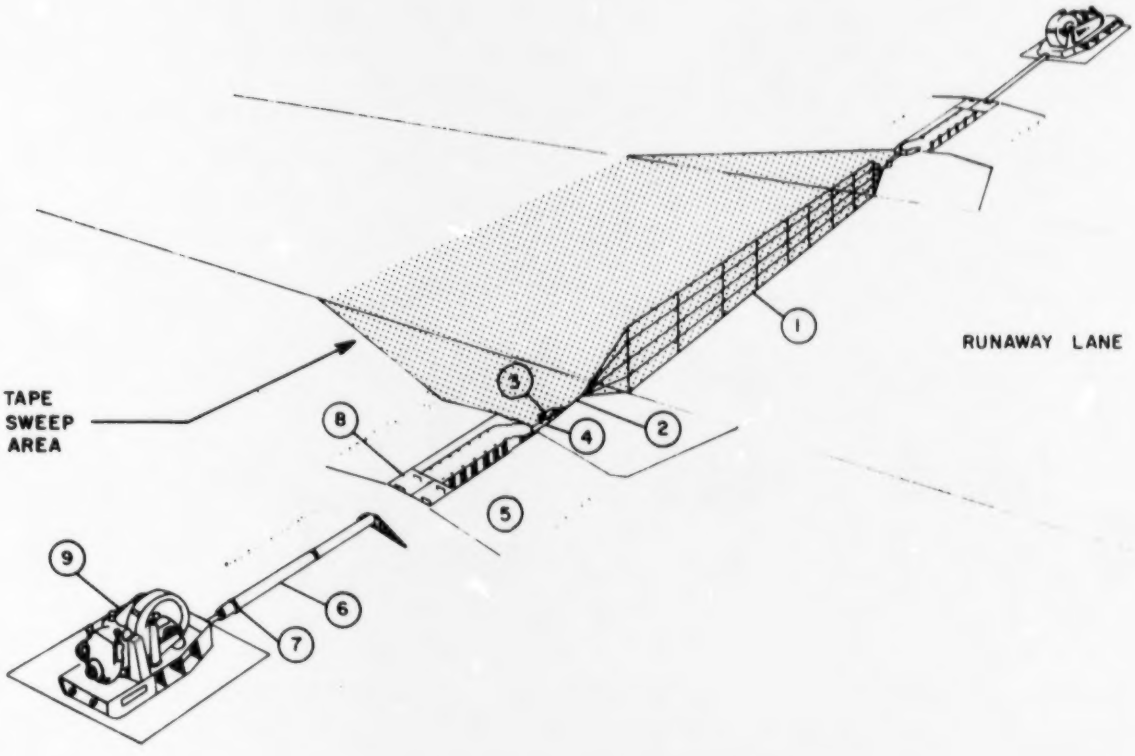
TABLE 4-2a
COMPONENT IDENTIFICATION
ARRESTER ENGINE
REWIND SIDE
(Reference Figure 4-3)

KEY	COMPONENT
1	CHAIN GUARD
2	REEL ASSEMBLY
3	REWIND ENGINE
4	REWIND SPROCKETS AND CAM
5	ENGINE BASE
6	GAS TANK
7	RESERVOIR (OPTIONAL)
8	COOLANT PUMP (OPTIONAL)
9	COOLANT MOTOR (OPTIONAL)

TABLE 4-2b
COMPONENT IDENTIFICATION
ARRESTER ENGINE
CONTROL SIDE
(Reference Figure 4-3)

KEY	COMPONENT
1	HYDRAULIC SYSTEM
2	REWIND CLUTCH HANDLE
3	BRAKE COOLANT SYSTEM (OPTIONAL)
4	CHAIN GUARD
5	TAPE LEAD-OFF SHEAVE
6	INSTRUMENT PANEL
7	BATTERY

A single arrester engine system is shown in Figure 4-4 and Table 4-3. A dual engine installation is shown in Figure 4-5 and Table 4-4. The operation is essentially the same for all configurations.

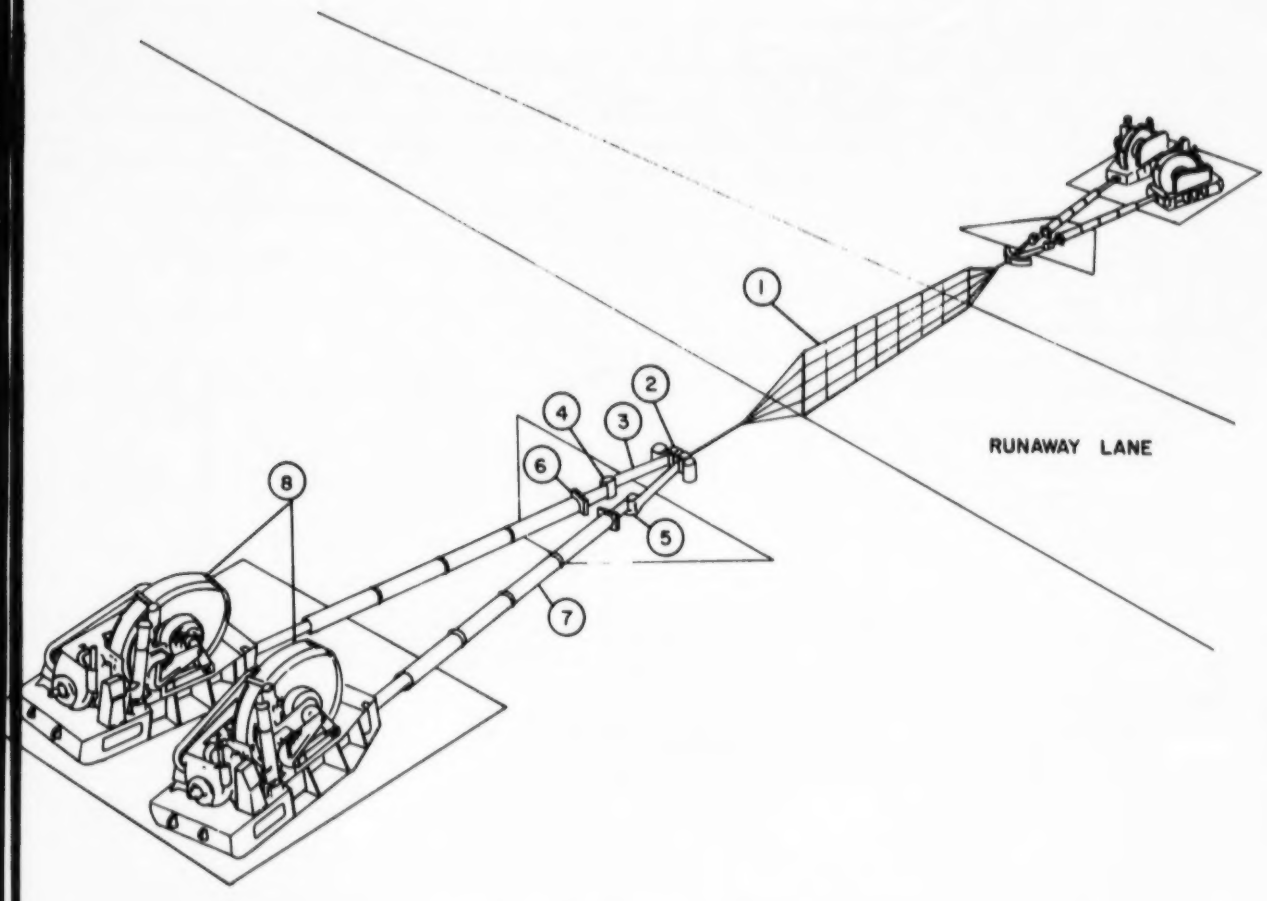


SINGLE ARRESTER ENGINE INSTALLATION

FIGURE 4-4

TABLE 4-3
COMPONENT IDENTIFICATION
SINGLE ARRESTER ENGINE INSTALLATION
(Reference Figure 4-4)

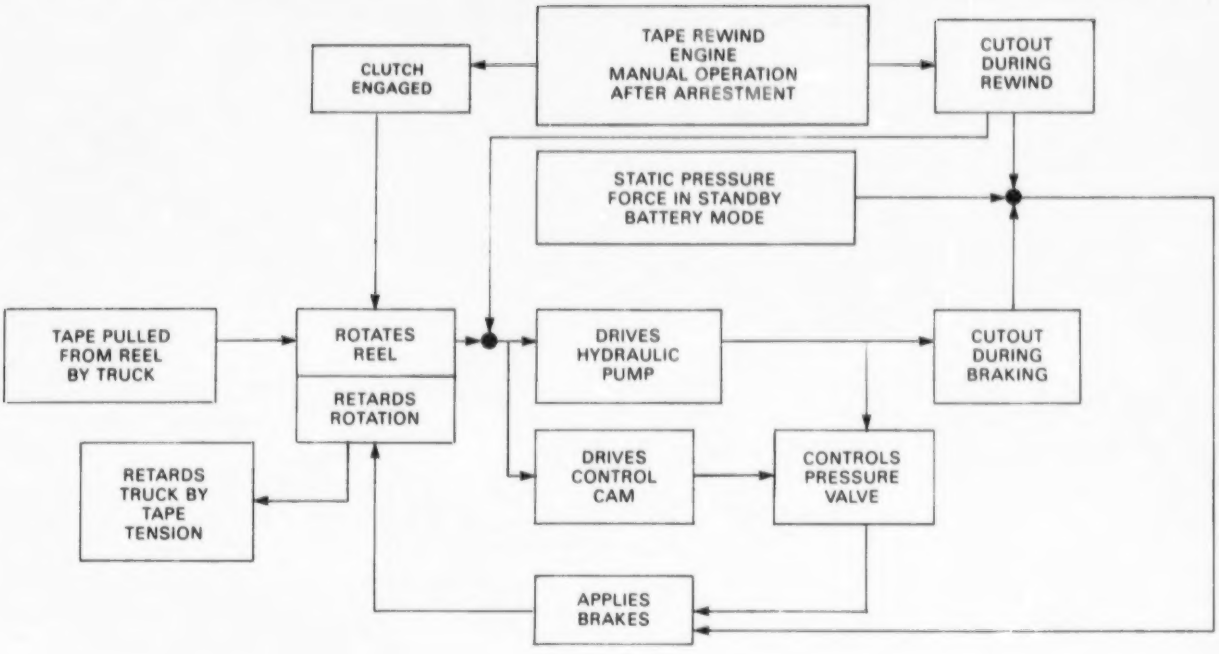
KEY	COMPONENT
1	ARRESTMENT BARRIER ASSEMBLY
2	BARRIER CABLE
3	TAPE CONNECTOR ASSEMBLY
4	PURCHASE TAPE
5	FAIRLEAD BEAM
6	FAIRLEAD TUBE AND COUPLINGS
7	FAIRLEAD TUBE ANCHOR
8	FAIRLEAD LEAD-ON SHEAVE
9	ARRESTING ENGINE



DUAL ARRESTER ENGINE INSTALLATION
FIGURE 4-5

TABLE 4-4
COMPONENT IDENTIFICATION
DUAL ARRESTER ENGINE INSTALLATION
(Reference Figure 4-5)

KEY	COMPONENT
1	ARRESTMENT BARRIER ASSEMBLY
2	TAPE CONNECTOR
3	PURCHASE TAPE
4	DECK SHEAVE ASSEMBLY
5	DECK SHEAVE ASSEMBLY (OPPOSITE HAND)
6	TAPE BRUSH ASSEMBLY
7	FAIRLEAD TUBES AND COUPLINGS
8	ARRESTING ENGINE



ARRESTER ENGINE OPERATION

FIGURE 4-6

System operation is diagrammed in Figure 4-6.

In standby, the system is in the battery mode. With the system in this mode, the runaway truck engages the barrier. As the barrier is pulled along the runaway lane, the purchase tape is pulled out and the tape storage reel is turned. The spinning storage reel drives a hydraulic pump. The pump pressure and flow is regulated by a cam on a control valve. In this manner the hydraulic force on the brakes is scheduled by the fluid flow from the control valve. The resulting braking force is applied to the tape storage reel by two sets of disk brakes, one on each side of the reel. The brake applied torque on the reel causes tension in the tapes which slows and stops the runaway vehicle. The barrier is then disengaged from the truck. The truck is removed from the runaway lane. The barrier and purchase tapes are inspected, repaired or replaced as wear dictates. The tapes are rewound onto the tape reels by the gasoline powered rewind engines. The barrier is reset and the system returned to battery mode to wait for the next runaway truck arrestment.

The major components of the arrester engine are the tape reel, disk brakes, hydraulic system, retraction system, brake cooling system, synchronizing system and the tape lead-off sheave. These components are supported on a welded steel frame. The frame, for this application, is mounted onto a reinforced concrete foundation.

4.1.1. Reel Assembly

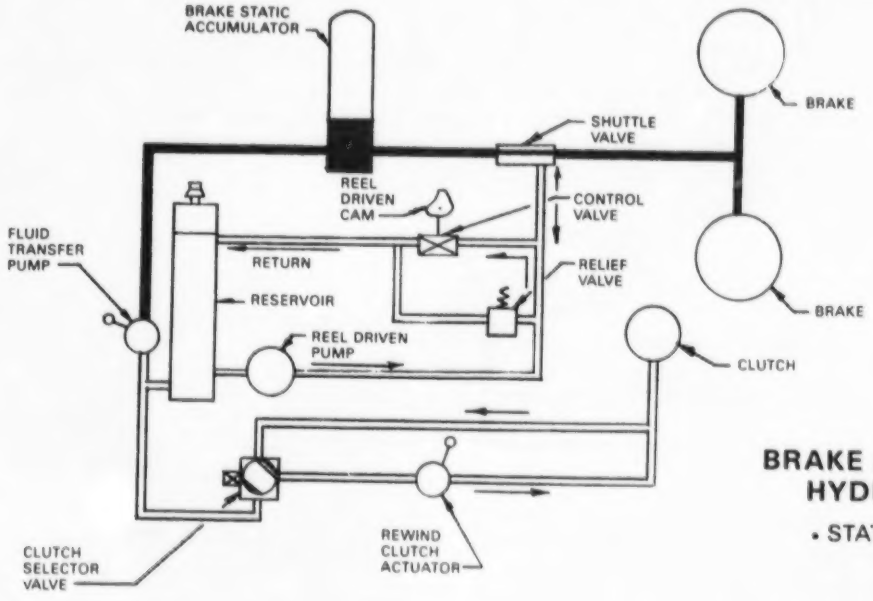
The reel assembly is the tape storage reel mounted on a shaft along with two sets of aircraft type disk brakes and a multi-disk clutch. The clutch is used to engage the retraction engine. The shaft rotates in self-aligning bearings mounted on each end of the assembly.

Mounted on the opposite end of the shaft to the retraction engine clutch are two sprocket assemblies. One sprocket drives the hydraulic system and the other, through a reduction gear, the control valve cam.

4.1.2. Hydraulic System

The hydraulic system is a velocity-position sensitive control system. It is powered by a reel driven pump and regulated by a cam actuated control valve. The system provides the capability of a wide range of weight settings with one basic cam contour. On current models low-to-high weight settings are selected by positioning either a lever linkage or a needle valve. The linkage lever adjusts the cam operating mechanism fulcrum point to either tighten or loosen the valve orifice. The needle valve can be pre-set to any of a number of orifice settings which control the pressures maintained by the control valve. In either case, the system weight selection must be made manually prior to system operation.

A static pressure system (Reference Figure 4-7) consisting of an accumulator, a fluid replenishing hand pump and a reservoir provides a light brake pressure on the reel to maintain tape tension when the system is in battery position. The static pressure system is isolated from the main hydraulic system by a shuttle valve. This shuttle locks out the static system whenever the main pressure system takes over the control during arrestment and during tape rewind to eliminate brake drag load on the reel. Pressure is provided in the static pressure system by the accumulator. Hydraulic fluid level is maintained by transferring fluid from the main hydraulic system to the accumulator by the hand pump. The static system pressure is controlled by charging the accumulator with nitrogen.

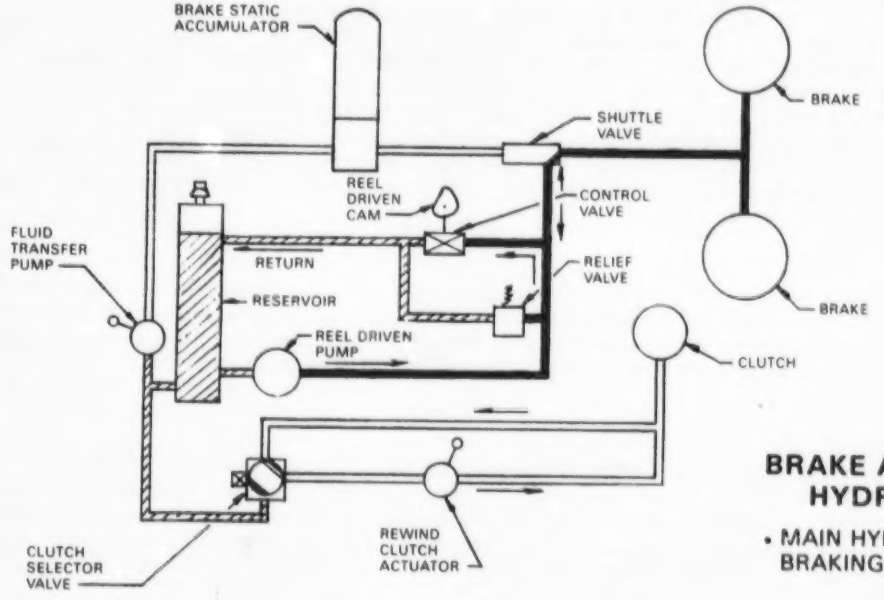


BRAKE ARRESTER ENGINE HYDRAULIC SYSTEM

• STATIC PRESSURE MODE

FIGURE 4-7

The main hydraulic braking system (Reference Figure 4-8) is powered by a hydraulic pump which is chain driven directly by the tape reel. The fluid level is maintained by a four gallon reservoir. Fluid is drawn from the reservoir by the pump and delivered to the return side through the cam controlled valve. As the pressure is built up and overcomes that of the static pressure system the shuttle valve moves to isolate the static system and allow the pressure built up by the pump to actuate the brakes. A relief valve is located in the manifold to prevent the braking system pressure from exceeding 13,790 ± 690 kilopascals. A balancing orifice is provided so that the two braking engines can be properly synchronized.

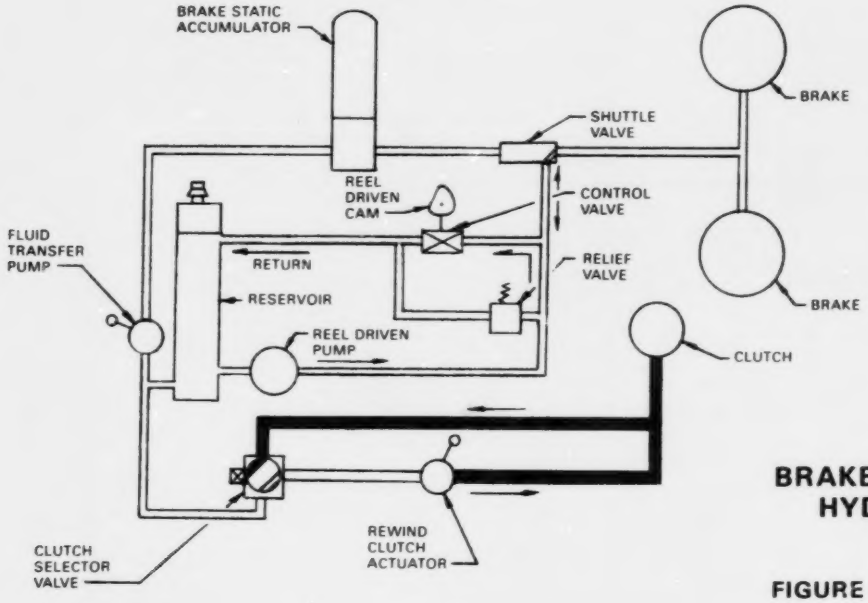


BRAKE ARRESTER ENGINE HYDRAULIC SYSTEM

• MAIN HYDRAULIC BRAKING MODE

FIGURE 4-8

The rewind clutch system (Reference Figure 4-9) consists of an actuating pump, a selector valve and a friction clutch. The hand-operated actuating pump stores fluid in its cylinder bore, and when actuated supplies this fluid under pressure to engage the rewind clutch. Under battery and arresting conditions the selector valve isolates the rewind clutch system. During clutch engagement, the selector valve is ported to the reservoir to supply fluid to the actuating pump. When hand pressure is released on the pump handle the pressure is released and equalized with the ambient pressure of the reservoir which is maintained by a breather filler cap.



BRAKE ARRESTER ENGINE HYDRAULIC SYSTEM

• REWIND MODE

FIGURE 4-9

4.1.3. Retraction System

The retraction system is used to rewind the purchase tape onto the tape storage reel after an arrestment and essentially consists of an air cooled gasoline engine, a fluid coupling, a reduction gear and a clutch. Several different engines are available on current production arrester systems. A detailed engine description is not presented as it does not contribute either to the understanding of the brake arrester engine system or its operation. It is suffice to say that the retraction engine is totally self contained including necessary fuel, carburetion, choking, starting, electrical, ignition, lubrication, governor, control and instrument systems. The retraction system operation is manually controlled.

4.1.4. Arrester Engine Synchronization

In order for the brake arrester system to function properly the two arrester engines must be synchronized so that both purchase tapes are pulling with equal force. To prepare the arrester engines for synchronization, typically the normal reel-to-pump and reel-to-cam drive chains are removed and a chain is installed between the braking engine drive and the hydraulic pump. In this configuration, the rewind engine is used to drive the pump. The objective is to equalize the hydraulic system output pressures for each of the two arrester engines. It is accomplished by obtaining a specific brake pressure at a reference location on the cam where the hydraulic pump is driven at a given speed. Specific synchronizing details are different for each engine model. Because of the heat generated during synchronization and operator involvement, a fire extinguisher must be on hand and the adherence to fire precautions strictly enforced.

4.1.5. Cooling System

An optional water spray brake cooling system is available for each model arrester engine. It is turned on during the first minute of the rewind cycle only when high frequency high energy arrestments are made. When the arrester engine is used for single arrestments on an emergency basis the brake cooling system is not required. It is anticipated that it would not be a requirement for the brake arrester engines when used for the arrestment of a runaway truck.

4.1.6. Fairlead Beam Assembly

The fairlead beam assembly consists of an alignment sheave, two tape brushes and the fairlead beam. The sheave is mounted on the front of the beam to maintain purchase tape alignment. The two brushes are installed behind the sheave to clean the tape of foreign material during the rewind operation so as to maintain a practical tape life. Behind the brushes the tape is turned through a 90 degree twist to line up with the horizontal axis of the tape reel. The tape, after the 90 degree twist, passes through the lead-on sheave and the fairlead tubing to the tape storage reel.

4.1.7. Fairlead Tube

The fairlead tube is only necessary for installations where the fairlead beam cannot be bolted directly to the arrester engine bed due to site layout restrictions. The tubing protects the tape from deterioration by sunlight, abrasion from wind flapping, possible vehicle damage, and against accumulation of mud, snow or wet sand which could damage or corrode arresting system components. The tubing is 10-3/4 inch (27.3 cm) OD by 3/16 inch (.5 cm) wall steel tubing joined by welded couplings. Grading is maintained by covering the tube with earth.

4.2 Maintenance and Inspection Requirements

In order to keep the arrester system in peak operational condition a continuous and conscientious maintenance and inspection program must be conducted by dedicated and trained personnel. As well as periodic inspections, the operating personnel must check the system after each arrestment and during every rewind. Any abnormal and erratic operation is cause for a major inspection to identify and repair the trouble. In addition to routine periodic maintenance and inspections, the arresting system should be overhauled every five years. To give a feeling for the extent of maintenance and inspection work involved, the recommendations made by the manufacturer for systems in operation at airports are presented in Tables 4-5 and 4-6. The truck arrester is not subject to the same degree of criticality or anticipated usage as an aircraft arrester. Therefore, it may be that the inspection periods may be extended, within limitations of manufacturers' warranties, as a cost saving. Daily inspections may be extended to a semi-weekly or a weekly basis and weekly inspections extended to a bi-weekly or a monthly period, and so forth. Only after actual site experience is gained could this policy be examined.

TABLE 4-5
PERIODIC INSPECTION SCHEDULE

INSPECT OR CHECK	FREQUENCY			
	DAILY	EACH ARREST'T	WEEKLY	MONTHLY
1 For loose or worn system parts and security of anchorage		X	X	
2 Fittings and lines for leaks		X	X	
3 Deck sheave and tape sweep areas for debris/obstructions	X	X		
4 All sheaves for freedom of rotation		X		X
5 Barrier cables for kinks, broken wires, wear and tension	X	X		
6 Drive chains for cleanliness and lubrication		X	X	
7 Tape for localized wear		X	X	
8 Tape connectors for correct reeving, hardware security, cracks, distortion and deformation		X	X	
9 Tape brushes and adjust for wear				X
10 Tape guides for wear				X
11 Brakes for wear				X
12 Reel assembly for wobble during operation		X		X
13 Fluid couplings for operation during rewind and pretensioning				X
14 Battery liquid level			X	
15 Fuel and oil quantity	X	X		
16 Operate rewind engine (10-15 minutes)	X			
17 Gages for "0" setting and reset as necessary	X	X		
18 Accumulator for fluid level and static pressure	X	X		
19 Fluid level in reservoir	X	X		
20 Cam for "0" index setting	X	X		
21 Rewind engine fuel filter and air cleaner			X	
22 Bleed hydraulic system		X	X	
23 System for corrosion		X		X
24 Control valve linkage		X	X	
25 Clearance of tape edge with side plates for wear		Each 5		X
26 System for rodent and animal damage	X	X		

TABLE 4-6
PREVENTIVE MAINTENANCE SCHEDULE

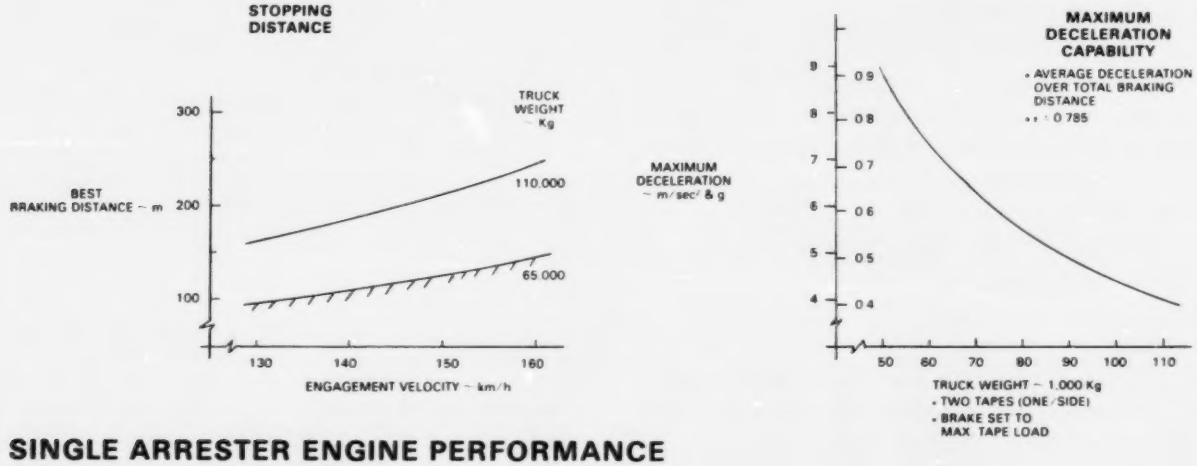
FUNCTION	FREQUENCY
1 Replace barrier	Every 24 months
2 Crop end of tape	Every 6 months
3 Turn tape end for end	Every 18 months
4 Synchronize units and proof test hydraulic system	Every 3 months
5 Functional checkout	Monthly
6 Clean entire equipment	As required
7 Replace tape	Every 48 months (or 64 arrestments)
8 Change oil filter	Every 90 days
9 Clean air filter	Every 6 months
10 Lubrication interval for all equipment	Every 30 days (Some every 90 days)

4.3 Arrester Engine System Estimated Performance

It is concluded that the arrester engine braking system can be modified for application as a runaway truck arresting system. The cam profile will have to be reshaped and the weight calibrating orifice/fulcrum reset to effect the system performance. As an example, the purchase tape ultimate strength is 47,628 kg. Tape life is 4 years or 64 arrestments. If both criteria coincide, life is equivalent to 1.33 arrestments per month. At any site, there will never be anything approaching this activity - at least from runaway trucks with failed brakes. Therefore, for properly maintained purchase tapes, it is safe to maintain a high design tensile load, say with a working factor of $\eta = 0.6$. Hence, the maximum tape load is 28,600 kg and the decelerating force on the truck is 57,200 kg for the single arrester engine configuration. The resulting deceleration is 5.10 m/sec^2, or 0.52 g, for a 110,000 kg heavy hauler and 8.60 m/sec^2, or 0.88 g, for a 65,000 kg truck. Corresponding stopping distances are 194 m and 115 m for the 110,000 and 65,000 kg vehicles respectively. These values are optimistic as they are based on the maximum braking force applied for the whole deceleration period, a most improbable condition. To allow for acceleration buildup and tailoff, an elliptical distribution with an ellipticity form factor of $\epsilon = 0.785$ is considered. The average deceleration is reduced to 4.00 and 6.75 m/sec^2 for the 110,000 and 65,000 kg trucks respectively. The associated stopping distances are increased to 247 (say 250) m for the 110,000 kg vehicle and 146 (say 150) m for the 65,000 kg truck. These data are presented in Figure 4-10 and represent a best effort case.

If the weight calibrating orifice/fulcrum is set at the heavy truck setting, the light vehicles undergo unacceptably high deceleration levels. Levels that exceed 10 g. In order to accept light vehicles, a feedback loop must be incorporated to automatically establish the weight setting. This would allow brake pressures compatible with the runaway truck size.

Based on the assumption that occasional use allows high tape loads, within tape life criteria, a single arresting engine system is expected to just handle a 110,000 kg heavy hauler truck. Comparing the energy levels shown in Tables 4-1 and 4-2, the energy absorbing capability of the 168 cm reel is adequate with a margin 5.5 per cent. The complications and expense of a dual arrester system do not appear warranted for the truck arrester application.

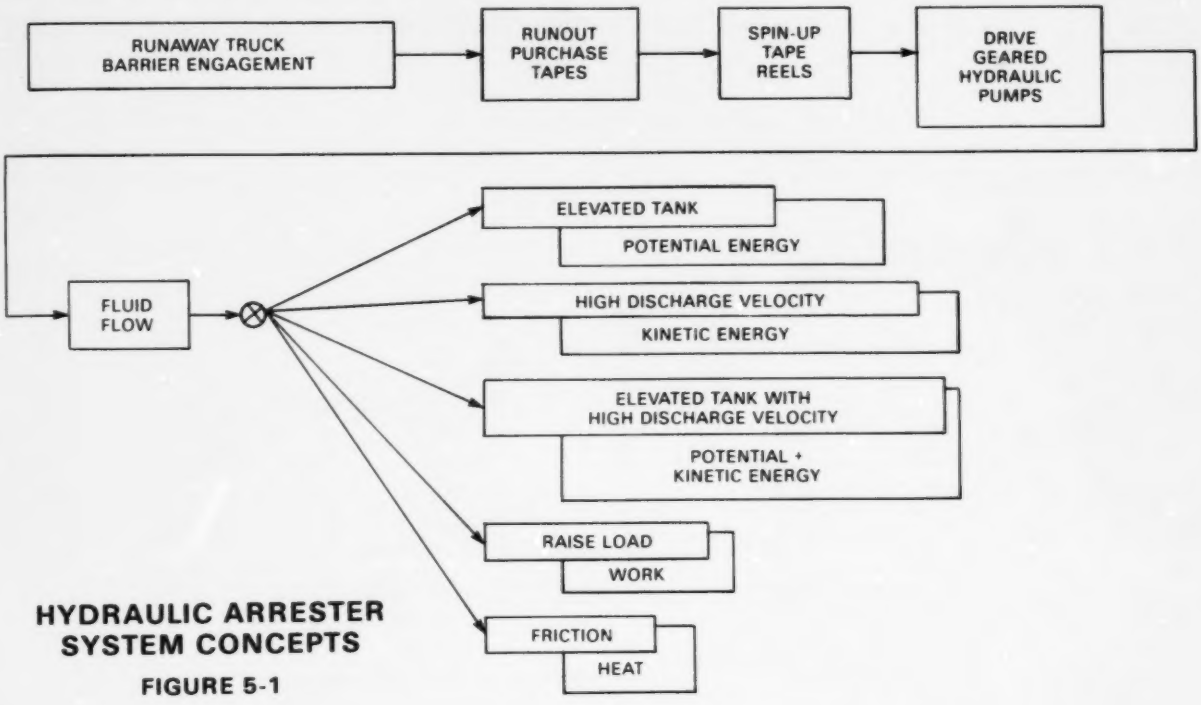


SINGLE ARRESTER ENGINE PERFORMANCE

FIGURE 4-10

5.0 Hydraulic Arrester System

Hydraulics, because of the apparent ease of converting from one energy level to another, is examined in this section to establish the technical feasibility of its application to runaway truck arrestment. The concept consists of an arresting barrier, purchase tapes, sheaves, fairleads and tape reels, as described in previous sections, centered about the runaway lane and driving geared hydraulic pumps. The pumps convert the kinetic energy of the runaway truck to other states of energy. Five concepts of hydraulic system arresters of different energy conversion methods, reference Figure 5-1, are looked at in the following sections.



HYDRAULIC ARRESTER SYSTEM CONCEPTS

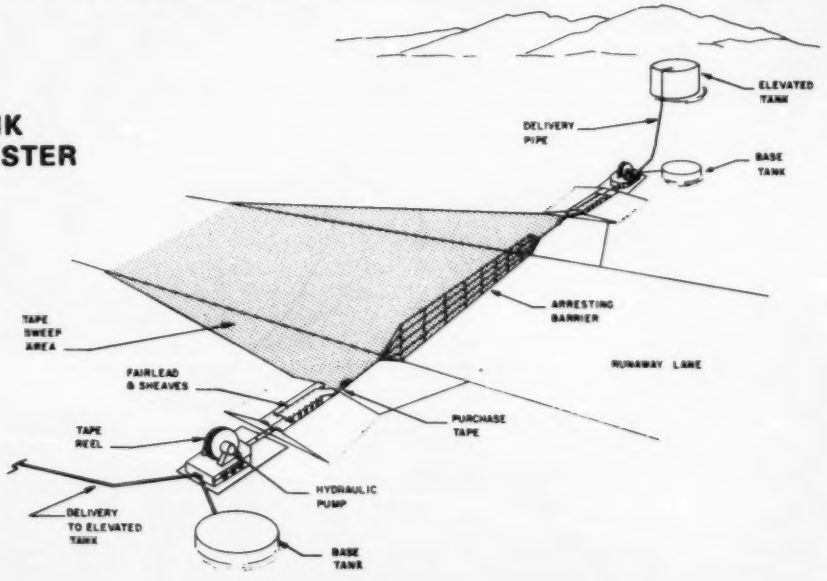
FIGURE 5-1

5.1 Elevated Tank Concept

The kinetic energy of the runaway truck is converted to potential energy by pumping a liquid from a base level tank into an elevated tank. The proposed concept is shown in Figure 5-2. The hydraulic pumps are driven with geared rotating tape reels which are spun-up by the purchase tapes being pulled out when the runaway truck is caught in the arresting barrier. Like the previous systems, the fairlead and sheaves turn, clean and direct the purchase tapes. The arrester reel and pump system is mounted on a steel weldment base and concrete pad. The hydraulic fluid is stored in base level storage tanks and is pumped to elevated tanks during the arresting operation. The fluid is retained in the elevated tanks by check valves until bypassed by manually operated valves. When bypassed, the fluid returns to the base level tank under gravity feed.

ELEVATED TANK HYDRAULIC ARRESTER

FIGURE 5-2



Assuming the fluid is an oil with a specific gravity of 0.85 and the design limiting case is the heavy hauler with a kinetic energy of 109 (10^6) Joules, at an engaging speed of 160 km/h, the potential energy of the pumped oil is 54.5 (10^6) Joules for each tank. Resulting physical system characteristics are shown in Figures 5-3 and 5-4 in relation to the head pumped. In Figure 5-3 the individual tank volume, based on two tank system, is shown by curve VOL. The tank dimensions are presented in curve DIA/HT for a tank with the height (HT) equal to its diameter (DIA). Another interpretation is shown in curve DEP which represents the oil depth in a large single cistern of 30 m diameter. The flow to each of the two tanks is shown in Figure 5-4, based on an average of 9.06 seconds deceleration time.

The conclusion is that even with two tanks elevated at 30 m the tank volumes are extremely large and the flow rates correspondingly high. Therefore, conversion of the truck's energy to fluid kinetic energy rather than potential energy is considered next as an alternative concept.

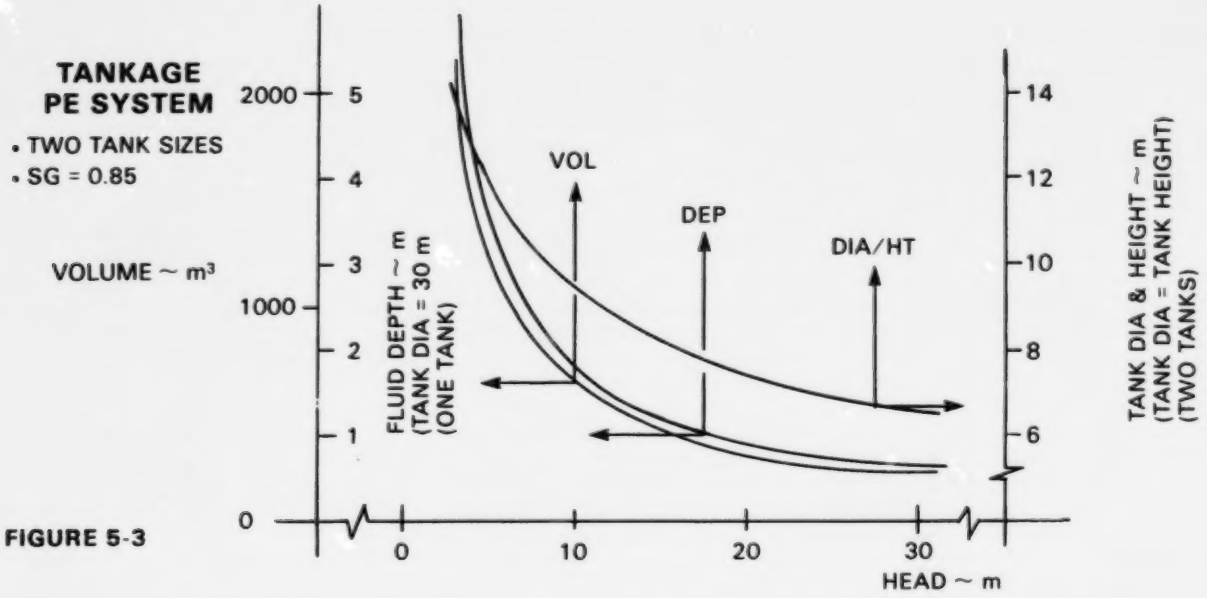


FIGURE 5-3

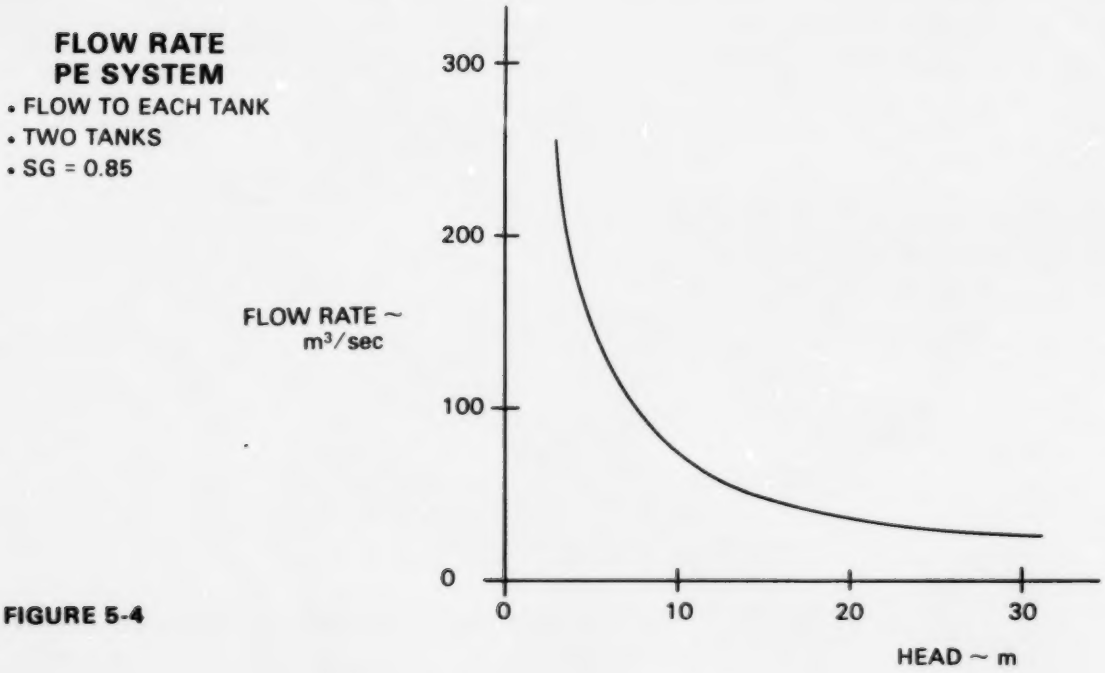


FIGURE 5-4

5.2 High Discharge Velocity Concept

Considering the case where all of the truck's energy is converted into kinetic energy of the fluid, the pumps discharge the oil through nozzles into cisterns at base level in a closed loop system. A system diagram is shown in Figure 5-5. A double cistern with a return pump is shown. An alternative concept would be to have one cistern with the pump inlet at the bottom of the cistern and discharging at the top. This configuration could have performance limitations due to fluid temperature rise and frothing. The data presented in Figures 5-6 and 5-7 are for the system concept of Figure 5-5. Again, the oil specific gravity is 0.85 and the design energy level is that of the heavy hauler with an engaging velocity of 160 km/h. For perfect conditions, the delivery pipe diameter and discharge pressure are shown in Figure 5-6. The volume of each tank, four required, and tank dimensions based on tank height equal to tank diameter are shown in Figure 5-7. It is observed that the tank size for the potential energy system with a head of 30 m is the same as that of the kinetic energy system with a discharge velocity of 24 m/s; i.e., the effect of the velocity squared term is very pronounced. Even so, it is clear that the system is not practical for the design energy level, in that the mass flow rate is just too much.

Therefore, it is concluded that a kinetic energy system, like the potential energy system, will not handle the design requirements in a practical manner. Hence, a combination of the two energy systems is considered next.

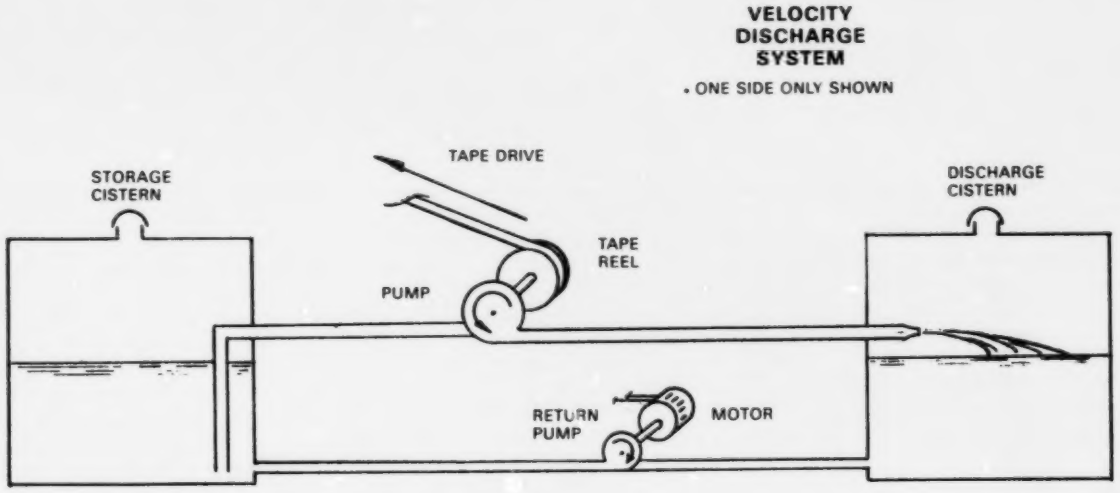


FIGURE 5-5

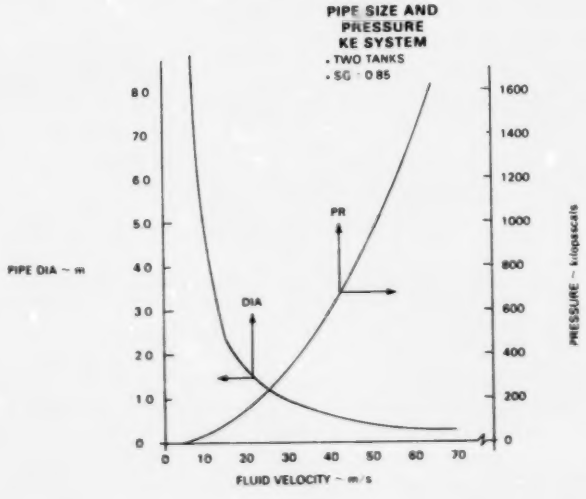


FIGURE 5-6

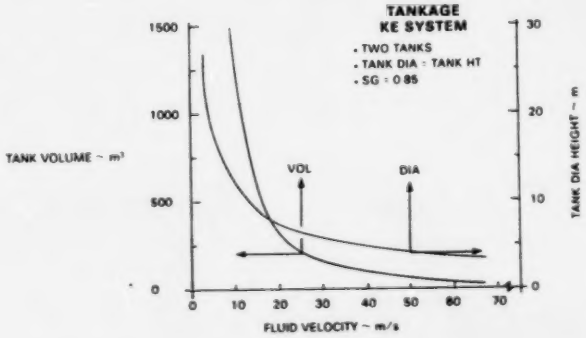


FIGURE 5-7

5.3 Elevated Tank with High Discharge Velocity Concept

The combined potential and kinetic energy system is conceptually shown in Figure 5-2. In this case, the fluid is pumped to the elevated tanks and discharged into them through nozzles at high velocity. System characteristics for the same energy level and oil as in the two prior cases are presented in Figures 5-8 and 5-9.

TANKAGE
PE/KE SYSTEM
- TWO TANKS
- DIA = HT
- SG = 0.85

TANK DIA & HEIGHT ~ m

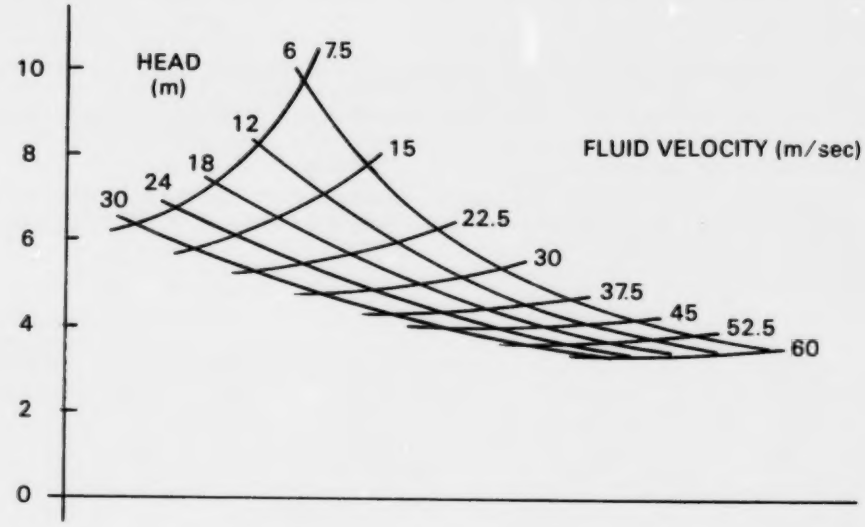


FIGURE 5-8

DISCHARGE
PIPE SIZE
PE/KE SYSTEM
- TWO TANKS
- SG = 0.85

PIPE DIA ~ m

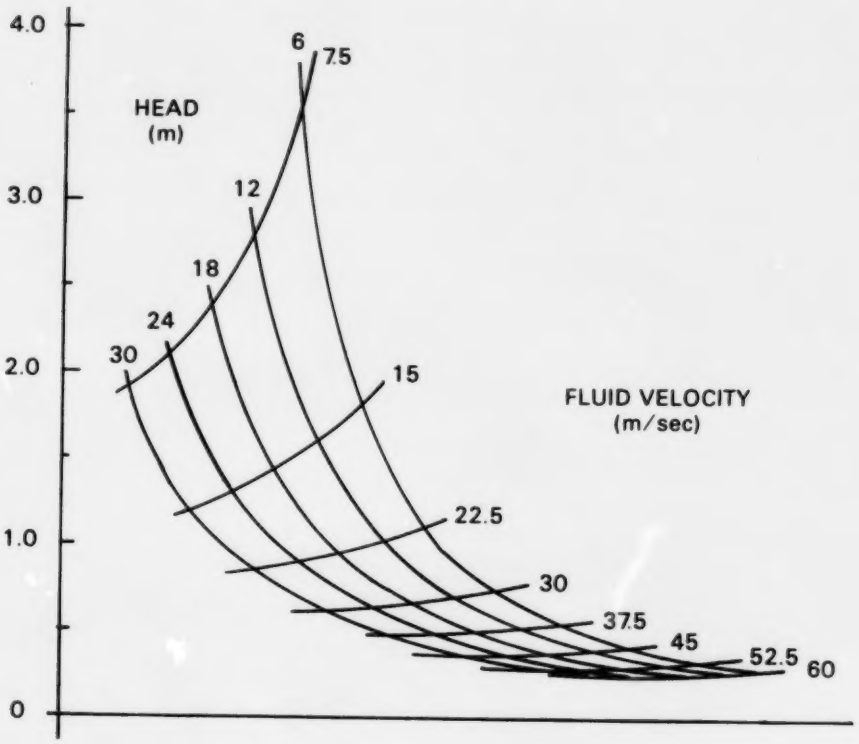


FIGURE 5-9

To scope the size of the system the tankage dimensions, with the tank diameter equal to the height, and the discharge pipe diameter are shown in relation to the head and discharge velocity. It is observed that the potential head is very influential at low discharge velocities but has very little effect at high discharge velocities. Even with the 30 m head and a 60 m/sec discharge velocity the tankage volume and mass flow are too large to represent a practical system based on the energy levels required to arrest the heavy hauler transporter. With only one pump on each reel, the size of each pump required is 6,000 KW (or 8,000 HP) - at least an order of magnitude larger than a realistic size and certainly a major turbo-machine design and expense. In point of fact, the pump inertia is so large that the pump could not be brought up to speed within the time required to stop the runaway truck.

It is therefore concluded that a runaway truck arrester system based on hydraulic work is not feasible. Such a system however, could be conceived for light vehicle arrestment which represents the system capacity limit - a case of only minor interest with respect to the objectives of this study.

5.4 Dead Weight Ballast Concept

Assume that an hydraulic drive connects the tape reel to a dead weight drive. For an average deceleration of 0.5 g, the stopping distance is:

$$s = \frac{v^2}{2a} = \frac{(44.4)^2}{9.8}$$

$$= 202 \text{ meters}$$

and the deceleration time is

$$t = \left[\frac{2s}{a}\right]^{1/2} = \left[\frac{2(202)}{4.9}\right]^{1/2}$$

$$\cong 9.1 \text{ seconds.}$$

The force required to provide this deceleration is

$$F = ma = \frac{110{,}000(4.9)}{9.8}$$

$$= 55{,}000 \text{ kg.}$$

For this concept, the dead weight ballast force is applied for the full stopping distance. It must therefore travel the same distance as the vehicle, or, the work must remain constant. That is, for example, if the distance is halved the weight is doubled, and vice versa. Lifting the weight vertically is obviously not attractive because of the great height of pull required. A horizontal pull would necessitate guiding the ballast along a track with slippers. It is necessary that the slippers provide a large coefficient of friction as the braking force is the product of the friction coefficient and the normal ballast weight, $F = \mu W$, for a horizontal track. Rain water and winter accumulations of snow and ice seriously reduce the friction to a near zero value. Without the frictional force, by either design or ambient conditions, the deceleration would only be due to the conservation of momentum. For example, $M_1 v_1 = (M_1 + M_2)\, v_2$, so that if the ballast weight equalled the truck weight $M_1 = M_2$ then $v_2 = \frac{1}{2} v_1$. Obviously not a braking system. Therefore, the addition of hydraulic operated mechanical brakes to sustain consistent friction is desirable. The dead weight ballast concept is considered unattractive and herewith is discarded as impractical without any further consideration.

5.5 Friction Concept

As the hydraulic requirements are too great for the direct energy conversion systems and the dead weight system depends upon friction to work, a high friction concept is desired. To allow the hydraulics to work, a high pressure low volume system is proposed. So that tonnes of ballast are not dragged hundreds of meters to produce friction, consider a mechanical brake applied directly to the tape reel. In its simplest form the system consists of a brake band mounted on a drum on the tape reel. To prevent reel lock-up, the brake diameter is about three-quarters of the reel diameter and the brake shoe area is limited to about one square cm per 14 kg. The brake area is $\frac{55{,}000}{14} \cong 4{,}000 \text{ cm}^2$ and represents a very acceptable design. Even more efficient than the drum brake is the disc brake. With disc brakes installed on the tape reel shaft, and along with the rest of the system; e.g., barrier, tapes, fairleads and sheaves; this hydraulic system exhibits adequate operating performance. It represents a satisfactory hydraulic system concept and is in fact the arrester engine braking system described in detail in Section 4.

It is concluded that the arrester engine braking system is the only technically viable hydraulic arresting system of the candidate hydraulic systems studied. The less sophisticated hydraulic arrester systems are not practically attractive.

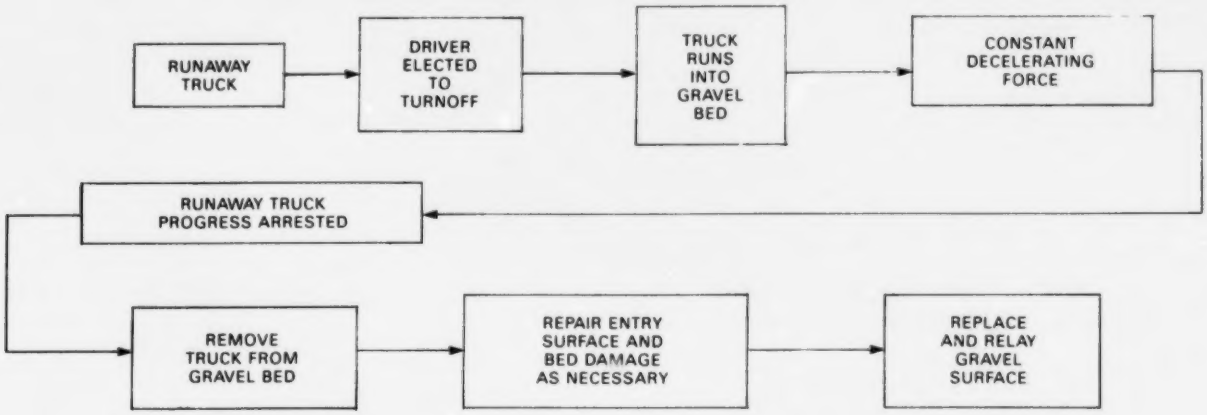
6.0 Passive Arrester System

All systems examined to this point in the study have had major system components as active braking elements. If these active elements could be eliminated resulting in a passive system, the non-recurring engineering and installation costs and the recurring operation and maintenance costs could conceivably be reduced. Elimination of the active members from the system leaves only the runaway lane. The braking coefficient may be controlled by careful selection of the runaway lane surface material. Selective material can be used; for example, sand, gravel, crushed aggregate, local soil, etc. The condition of the lane surface is important as a freshly turned surface will maintain a higher drag coefficient than a hard packed surface.

6.1 Gravel Bed Arrester System

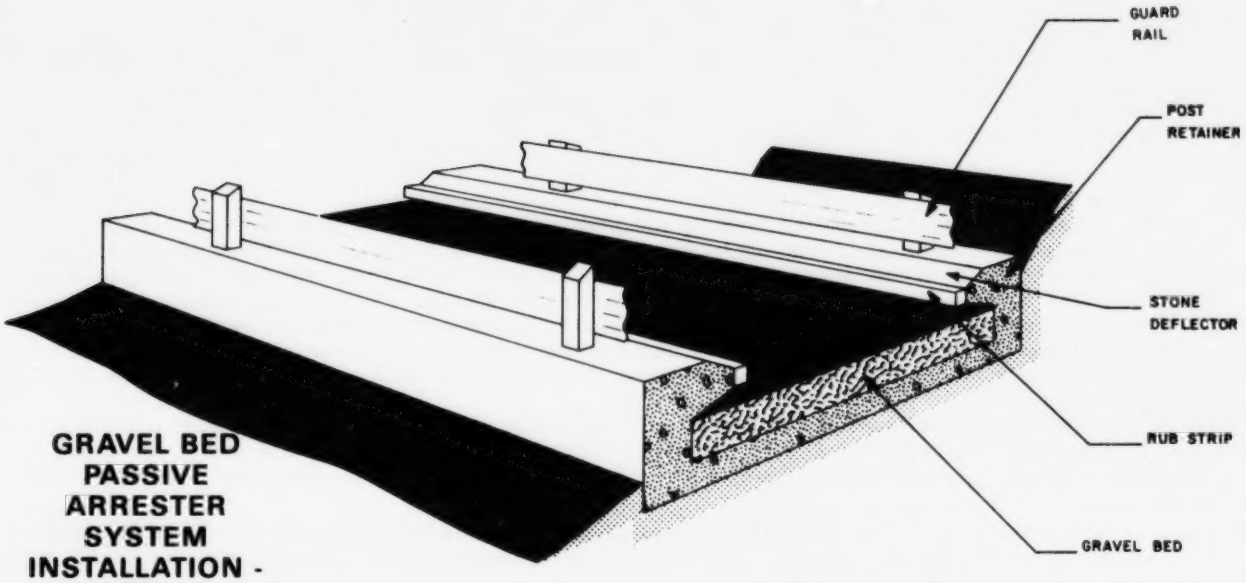
A pea gravel bed is one seriously studied and prototyped outside of Canada. Without the benefit of test data but assuming that the system must have some merit to warrant the capital investment of prototyping, a gravel

system was considered as a likely passive braking system concept. The operation of such a system is presented in Figure 6-1 and concept schematic shown in Figure 6-2. The basic concept of a passive system is certainly the simplest of the concepts studied. It consists chiefly of a runaway lane with a regulated depth of sized gravel contained in a form. A runaway vehicle upon entry into the gravel bed is decelerated by the frictional load of the wheels and vehicle undergear dragging in the loose gravel.


RUNAWAY TRUCK
DRIVER ELECTED TO TURNOFF
TRUCK RUNS INTO GRAVEL BED
CONSTANT DECELERATING FORCE
RUNAWAY TRUCK PROGRESS ARRESTED
REMOVE TRUCK FROM GRAVEL BED
REPAIR ENTRY SURFACE AND BED DAMAGE AS NECESSARY
REPLACE AND RELAY GRAVEL SURFACE

GRAVEL BED PASSIVE ARRESTER SYSTEM OPERATION

FIGURE 6-1



GRAVEL BED PASSIVE ARRESTER SYSTEM INSTALLATION - SECTION

FIGURE 6-2

Although test data of the gravel bed braking are not available, it was assumed that the friction coefficient for selected gravel is 0.35 with a range of from 0.25 to 0.40. From Newton's law,

$$F = ma$$

$$= \frac{W}{9.8} a$$

where the mass $m = \frac{W}{9.8}$ with W the truck weight in kilograms and a, the acceleration in m/sec^2.

The frictional braking force is

$F = \mu W$ where μ is the friction coefficient.

Substituting for F,

$$F = \mu W = \frac{W}{9.8} a$$

$$\text{or } \mu = \frac{a}{9.8}.$$

For acceleration in g,

$a = n\,g$

$= n\,(9.8).$

Substituting for a,

$$\mu = \frac{n\,(9.8)}{9.8}$$

or $\mu = n$;

i.e., the friction coefficient is equal to the deceleration when quoted in "g". It is noted that the deceleration is theoretically independent of vehicle weight. This is an important observation. It is the only braking system studied where light vehicles are treated in the same manner as a heavy counterpart. In other concepts the systems were not compatible with both light and heavy vehicles. If the other concepts were designed for the heavy hauler trucks, light duty trucks and private automobiles would be totally demolished when using the emergency runaway lane at high engagement speeds. The weight independence of the gravel bed passive concept is the single most important observation made in this exploratory design study and probably outweighs any advantages of the other conceptual systems. However, in practical cases, the braking will not be the same for all vehicles. The configuration will certainly be expected to influence the braking coefficient; viz., number of wheels, tire floatation, tire tread style, height of trailer bed, fender style, etc. Test data are required to establish the variation in braking coefficient with vehicle configuration and ambient conditions. Assuming that a 45 cm deep gravel bed was located on a level runaway lane for a constant braking coefficient of 0.35 and that braking is fully available upon entry into the gravel bed (t = 0 seconds), the braking performance was numerically calculated. The resulting system performance estimates are summarized in Figure 6-3. Data are shown for 145 km/h and 160 km/h entry speeds. As the deceleration is considered constant, the velocity decreases linearly. The distance to stop is under 300 metres for the worst case. The jerk is theoretically a snap forward at entry into the bed and backward at arrestment. Ideally, since the deceleration is constant and the braking coefficient $\mu = 0.35$, then

$$\frac{d^2x}{dt^2} = -(0.35)\,(9.80665)$$

$$= -3.4323 \qquad (m/sec^2)$$

Integrating,

$$\frac{dx}{dt} = -3.4323\,t + C_1$$

where $C_1 = V_0$ at $t = 0$

$= 44.4444$ m/sec (160 km/h)

then $$\frac{dx}{dt} = -3.4323t + 44.4444 \qquad (m/sec)$$

Integrating again,

$$x = -1.7162\,t^2 + 44.4444\,t + C_2$$

where $C_2 = 0$ at $t = 0$ so

$$x = -1.7162t^2 + 44.4444\,t \qquad (m)$$

As velocity reaches zero,

$$\frac{dx}{dt} = 0 = -3.4323\,t + 44.4444$$

or $t = 12.95$ seconds

and $x = -1.7162\,(12.95)^2 + 44.4444\,(12.95)$

$= 287.75$ meters, which check with the numerically calculated values presented in Figure 6-3.

The sensitivity of the braking coefficient is presented in Figure 6-4 for a range of $\mu = 0.25$ to 0.40. Over the range, the stopping time and distance are reduced by 40 per cent as braking coefficient is increased from 0.25 to 0.40. As the performance of Figure 6-3 was based on $\mu = 0.35$, a reduction in the coefficient to $\mu = 0.25$ would increase the stopping distance by 42 per cent whereas increasing the coefficient to $\mu = 0.40$ would decrease the distance by 13 per cent. The condition of the gravel bed is therefore very important.

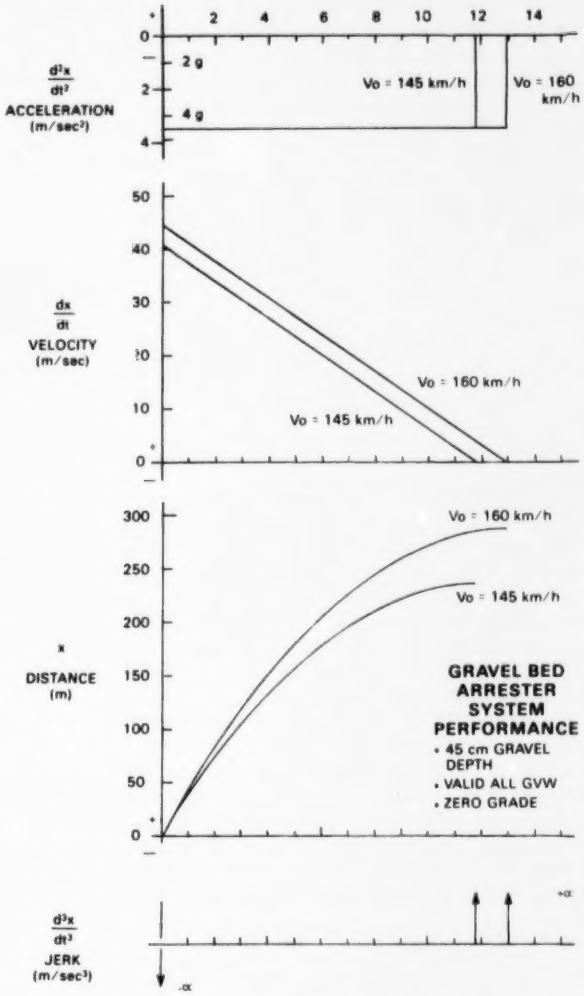


FIGURE 6-3

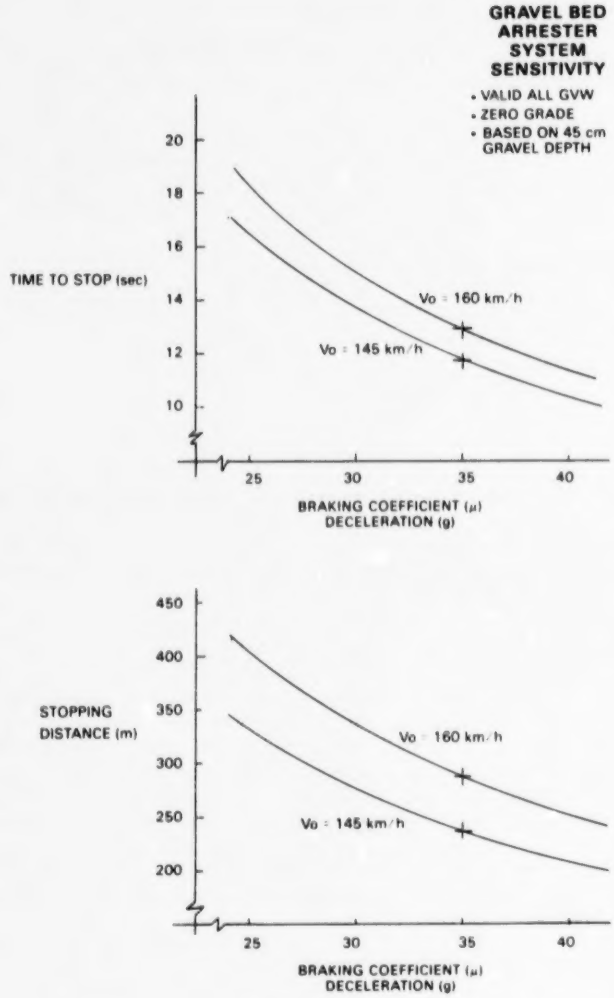

GRAVEL BED ARRESTER SYSTEM SENSITIVITY
• VALID ALL GVW
• ZERO GRADE
• BASED ON 45 cm GRAVEL DEPTH
TIME TO STOP (sec)
Vo = 160 km/h
Vo = 145 km/h
BRAKING COEFFICIENT (μ) DECELERATION (g)
STOPPING DISTANCE (m)
Vo = 160 km/h
Vo = 145 km/h
BRAKING COEFFICIENT (μ) DECELERATION (g)

FIGURE 6-4

6.2 Uphill Gravel Bed Arrester System

The performance summarized in Figure 6-3 is for a level (horizontal) runaway lane. If the lane, and gravel bed, are constructed with an uphill slope additional braking will occur due to lifting the vehicle weight. The drag component is

$$D = W \sin \theta$$

where θ is the grade angle in degrees.

In terms of a friction coefficient

$$\rho = \sin \theta.$$

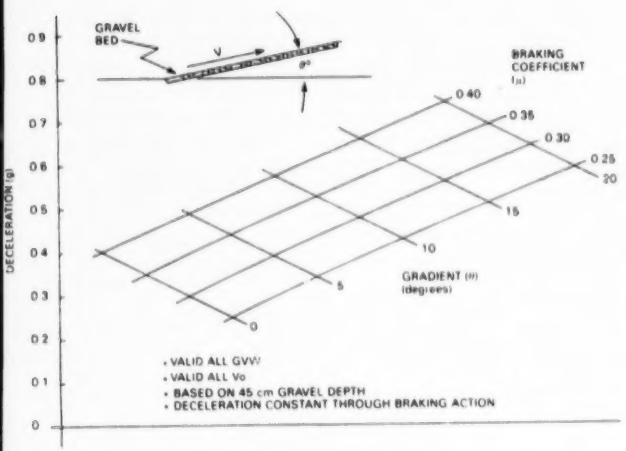
Numerical values are given in the following Table 6-1.

TABLE 6-1

GRADIENT INDUCED FRICTION COEFFICIENT		
GRADE θ (DEGREES)	FRICTION COEFFICIENT (ρ)	GRADIENT (%)
0	0	0
5	0.087	8.8
10	0.174	17.6
15	0.259	26.8
20	0.342	36.4

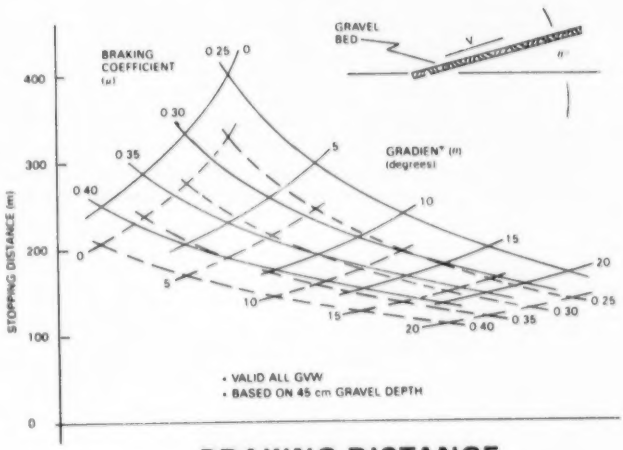
The total braking coefficient is the sum of the friction coefficients of the gravel bed and the gradient, $\mu + \rho$. For the uphill gravel bed arrester with the braking coefficient equal to $\mu + \rho$, performance was numerically calculated and is parametrically shown in Figures 6-5 through 6-7. The braking coefficient range for gravel of $\mu = 0.25$ to 0.40 and a gradient range of from $\theta = 0$ to 20 degrees are shown.

Figure 6-5 depicts the deceleration (g) and Figure 6-6 the stopping time. The stopping distance is given in Figure 6-7 for entry speeds of 160 km/h and 145 km/h. The gradient is most effective at low values of braking coefficient and, similarly, the braking coefficient is most effective at low gradients. Actual design compromises would depend upon topographical constraints of the sites selected. A general observation is that it appears desirable to shoot for as high a breaking coefficient μ as possible along with a moderate grade. The stopping distance histories are summarized in Figure 6-8 for $\mu = 0.35$ and an entry velocity of 160 km/h.



DECELERATION GRAVEL BED ON UPHILL GRADIENT

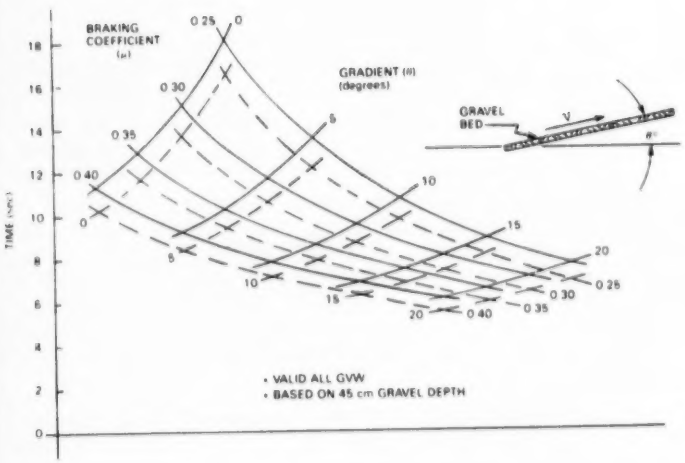
FIGURE 6-5



BRAKING DISTANCE GRAVEL BED ON UPHILL GRADIENT

- V_o = 160 km/h
- V_o = 145 km/h

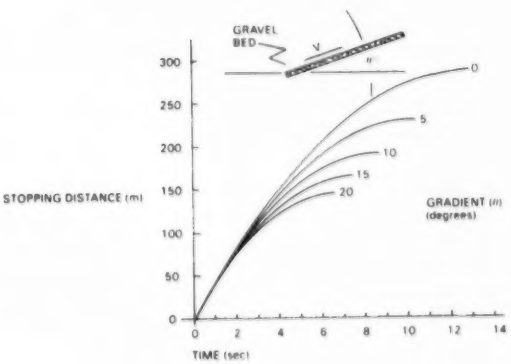
FIGURE 6-7



STOPPING TIME GRAVEL BED ON UPHILL GRADIENT

- V_o = 160 km/h
- V_o = 145 km/h

FIGURE 6-6



BRAKING DISTANCE GRAVEL BED ON UPHILL GRADIENT

- VALID ALL GVW
- BRAKING COEFFICIENT $\mu = 0.35$
- 45 cm GRAVEL DEPTH
- V_o = 160 km/h
- CONSTANT DECELERATION

FIGURE 6-8

6.3 Gravel Bed and Chain Arrester System

If the site is extremely restrictive an option is to incorporate a chain arrester system into a gravel bed design. The reduced stopping distances, compared to those of Figure 6-8, due to addition of a chain system are given in Figure 6-9 for gradients of 0, 5 and 10 degrees and a heavy hauler weight of 110,000 kg. Once the chain arrester is incorporated the system becomes vehicle weight sensitive again, eliminating the major advantage of the gravel bed. The information of Figure 6-9 is based on the assumption that all braking, both gravel and chain components, is applied at entry into the pit at t = 0 seconds. The stopping distances are attractive for the heavy hauler. For example, adding two chains to any of the base systems is about equivalent to a 10 degree gradient increase. The associated deceleration levels, for the heavy hauler, are from 0.35 to 1.12 g - acceptable levels. However, if a small vehicle of 2,000 kg runs into this setup the deceleration will exceed 5 g for two chains. The stopping distance is about 40 meters and the decelerating time lasts only about 1.3 seconds. A totally unacceptable condition - the light vehicles would be very severely damaged. The stopping would most likely be fatal to the driver and passengers.

To reduce this condition, the chain arrester could be located partway down the gravel bed. For example, setting the barrier 100 metres down the gravel bed results in estimated performance approximately equivalent to an overall gradient increase of 5 degrees for the heavy hauler. The peak deceleration is about 2/3 of a g. The peak deceleration is still over 4 g for the 2,000 kg light vehicle with the gravel bed at zero gradient and Vo = 160 km/h. The stopping time has increased to almost 4 seconds and the stopping distance increased to 130 metres. For barrier engagement at 100 metres, the engagement velocity is reduced from 160 km/h to 130 km/h. Similarly, the 7,000 kg vehicle undergoes a peak g of about 2.25 and is stopped in about 150 metres. Hence it is concluded that such a system would be expected to work for all but the lightest of vehicles.

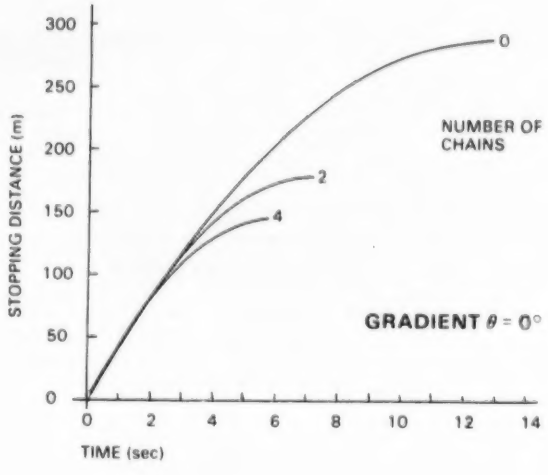


FIGURE 6-9 a

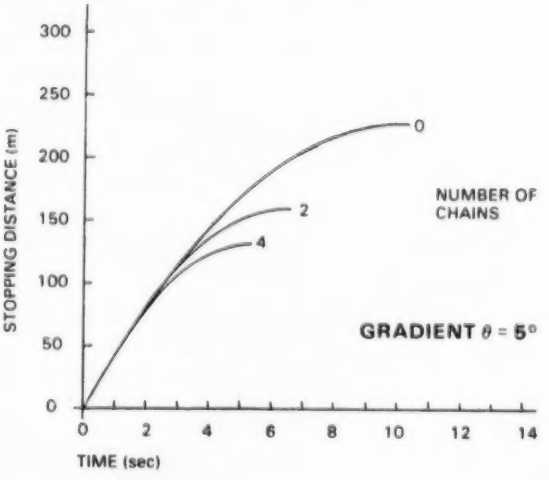
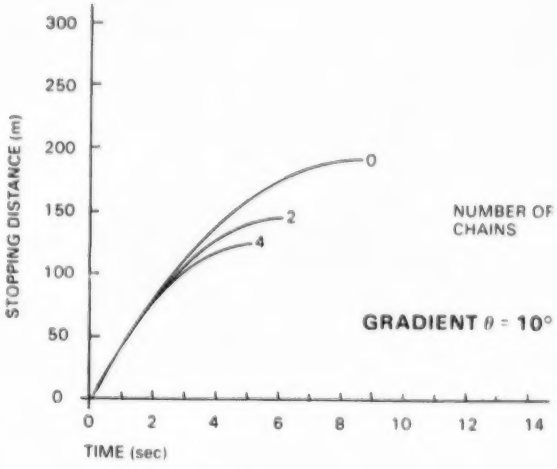


FIGURE 6-9 b

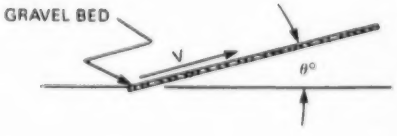
BRAKING DISTANCE GRAVEL BED WITH CHAIN ARRESTER



- GVW = 110,000 Kg
- BRAKING COEFFICIENT μ = 0.35
- 45 cm GRAVEL DEPTH
- Vo = 160 km/h
- ALL BRAKING STARTS AT t = 0 seconds

FIGURE 6-9 c

BRAKING DISTANCE GRAVEL BED WITH CHAIN ARRESTER



BRAKING DISTANCE GRAVEL BED WITH CHAIN ARRESTER

FIGURE 6-9

6.4 Discussion of Passive Arrester

Specific topographical characteristics of individual sites should be studied before the gradient and bed location are established.

It is likely that each site would have its own peculiar design. An alternative design consideration may be to cater for the lighter vehicles and let the heavy hauler at the highest entry speeds take the brunt of arrestment damage. Based on the assumption that the larger trucks are better maintained they may be consequently statistically less probable to have a runaway problem. Statistical truck runaway data should be accumulated and assessed to understand the probabilities of runaway.

Figure 6.2 is intended as a conceptual configuration only and not a hard design proposal. The concept is to have the gravel bed layed in a form. A stone deflector is incorporated along the top edges to return flying stones to the underside of the truck and the gravel bed. A rub strip and guard rail are incorporated to maintain directional control, a rubbing braking component and constraint against jackknifing. Maintaining directional control and preventing jackknifing is a design problem associated with all of the arrestment systems studied. It is concluded that rub strip and guard rail constraint is necessary in any and all alternative braking concepts. The gravel bed arrester therefore is seen, at least in its simplest form, as the starting point of all arrester systems. It is therefore the fundamental and most cost effective arrester of the alternatives studied.

Although the gravel bed is most appealing because of its apparent simplicity, there are several design problems that must be addressed;

- method of truck removal from bed
- relaying and maintenance of gravel
- drainage requirements and methods
- snow/ice accumulation, removal or prevention.

To shorten the runaway length, a terminal stopping component may be incorporated. For example, for a braking coefficient $\mu = 0.35$, entry velocity of 160 km/h and a level bed, the truck velocity is 25 km/h at under 200 metres distance. At this point a terminal stopping arrester can be added. Such an arrester could consist of:

- a sand (or similar material) pile
- a bed of shear posts
- frangible material wall
- crushable bulkhead (accordian compression)
- large plastic drums with loose tops and filled with a displacement material, for example water.

This terminal barrier could be located at least at the point where the velocity is expected to be equal to the regulated "crashworthy speed".

7.0 Barrier/Vehicle Catchment

All active systems require a method of catching the truck in a runaway lane. One possible method is the use of a barrier net to snare the speeding vehicle. Such a net was shown schematically in the figures for the various concepts in preceding sections of this report.

Barrier nets are fabricated from nylon and cotton textile webbing and are in service by the military for many purposes. The material is standardized and its characteristics are defined by military specifications. Some typical webbing and cordage materials are summarized in Table 7-1.

As indicated in this table, webbing and cordage are available in most any size and strength desired for the fabrication of a barrier net. Nylon is the preferred material because of its general weathering excellence and ability to stretch under load (about 30 per cent).

Arresting barrier nets were used in the military for aircraft emergency landing overruns. They have been taken out of service except for crash barrier use on aircraft carriers. The serious problem with the net was the excessive damage done to the airframe by the webbing. In many cases, the aircraft had to be totally written off. Many present day land based fighter airplanes have arrester hooks installed that are lowered to snare a cable stretched over the runway surface, à la carrier operated Naval aircraft. The cable is attached to arrester engines. A pop-up cable is sometimes used with aircraft not equipped with an arrester hook. The intent is for the cable to be caught on the main landing gear struts for load transfer.

For runaway truck protection, a requirement to install an arrester hook would likely be scorned and ignored. Legislating its installation would not be justified. The pop-up cable would have little success because of the many geometries and bed heights of trailers in common use. In many cases there is no major load carrying structural member for the cable to catch and remain fixed to either the tractor or trailer. The barrier net, for the same reason, would be a very dangerous apparatus. The only load carrying structure of the tractor is the front bumper, the main frame and possibly the engine mount. Much of the rest of the currently designed and manufactured unit is fiber glass, light gauge metal and glass. Barrier net induced damage to the vehicle would be severe. With the tapes possibly shearing through the tractor cab structure, the driver's life could be placed in jeopardy. Before a barrier net could be seriously considered much testing and net tailoring would have to be done.

Another shortcoming of the barrier net is that by catching a trailer rig at the front of the trailer the stopping force is applied at the wrong end of the vehicle. Force application at the front is destabilizing and will tend to induce jackknifing and overturning of the rig. On the other hand the arrester hook, if in fact it was installed at the back end of the trailer, ensures that the applied braking force is stabilizing and acts in a safe direction.

TABLE 7-1
MILITARY SPECIFICATIONS
FOR
WEBBING AND CORDAGE

SPECIFICATIONS	TYPE	SIZE (in)*	BREAKING STRENGTH (lb)*
WOVEN NYLON WEBBING			
MIL -W-4088H	XXVI	(1-3/4) x (.15-.18)	15,000
	XXIII	(1-1/8) x (.2-.3)	12,000
	XIX	(1-3/4) x (.1-.13)	10,000
	XXVII	(1-23/32) x (.085-.11)	6,500
	XXI	(1-1/4) x (.065-.085)	3,600
	IV	(3) x (0.25-.040)	1,800
	I	(9/16) x (0.25-.040)	500
TUBULAR NYLON WEBBING			
MIL-W-5625H		3/8	950
		1/2	1,000
		5/8	2,250
		1	4,000
COTTON WARP WEBBING			
MIL-W-5665H	I	(9/16) x (.040-.050)	350
	III	(1-1/4) x (0.40-.050)	750
	VII	(1-3/4) x (.140-.150)	2,600
	X	(1-3/4) x (.125-.170)	5,000
NYLON CORD			
MIL-C-5040F	I		100
	II		400
	III		550
	IV		750
CORELESS NYLON CORD			
MIL-C-7515E	XIII		50
	XV		200
	II		550
	IV		1,000
	VII		2,500
	X		5,000
	XII		10,000

*Specification Values

It is concluded that truck catchment by arrestment barrier or cable is technically the weakest component of the active arrestment systems. Much testing is required to design a satisfactory nondestructive and safe catchment system. It is considered a high development risk item.

8.0 Discussion and Recommendation

The objectives of this exploratory development study were met. Results show that a runaway truck arrestment system appears practical. By means of analysis the feasibility of some arrester systems was established whereas others failed and these concepts were rejected. Some definite conclusions were reached and recommendations put forward for follow-on refinement and preliminary design of a sound, safe and cost effective candidate system.

8.1 Discussion

None of the arrester concepts studied satisfied all of the Study Technical Specifications criteria. However, this is not to say that a realistic system was not indicated. Of the five basic arrester systems investigated only two will not work. These systems, the inertia wheel and the hydraulic, at best degenerated into the engine brake arrester.

- **Engine Brake Arrester**

The engine brake arrester system is a sophisticated, well proven and established production system. It can accommodate braking of the 110,000 kilogram heavy hauler but at that the manually selected setting will not accommodate light vehicles. It may be possible to design, test and incorporate an automatic feedback servo which will allow safe braking over the complete vehicle weight range. Periodic checking and balancing of the left and right engines is essential for safe operation and prevention of truck jackknifing and roll over. The engines are dependable but require a great deal of attention.

Specially trained and highly skilled service technicians are required for reliable engine brake system operation and its component maintenance and overhaul. The engine brake is by far the most expensive of the candidates to procure, install and maintain. An arrestment barrier is a necessary component of the engine brake system. A much less expensive alternative is the chain arrester.

- **Chain Arrester**
  Like the braking engine, the chain arrester cannot accommodate the complete vehicle weight range with one basic configuration; i.e., with a single fixed chain weight. However, once set up it requires very little maintenance and normally no materials. It remains, until used, in a static condition. Left and right chain drag forces must be identically balanced or lateral upsetting moments, which are naturally unstable, will be applied to the runaway vehicle causing jackknifing and overturning. This force imbalance cannot be eliminated but may be balanced out during operation. Initial procurement and operating costs are moderate. It is flexible to the degree that increasingly more pairs of chains can be added if a particular site allows only an exceptionally short runaway braking lane. However, this flexibility is at the expense of increased deceleration loads. The vehicle weight that can be safely stopped is consequently increased and the safe operating weight range becomes very restricted. An arrestment barrier is a necessary component of the chain arrester, as it is with the engine brake.

- **Arrestment Barrier**
  As pointed out, the two preceding active arrester sytems require some form of arrestment barrier to catch the speeding runaway truck. A single cable would be sufficiently strong to carry the braking and inertia loads but would apply unsafe point loads onto the vehicle. A barrier net would allow the load to be distributed over the truck body. But unfortunately most of the late model vehicles have fiber glass front structure and are not capable of carrying the applied high impact and shear loads of catchment and braking. It is felt that the barrier element of the active systems is extremely dangerous and a high risk development item. Elimination of the barrier would result in a passive system totally independent of potentially dangerous nets and expensive mechanical mechanisms.

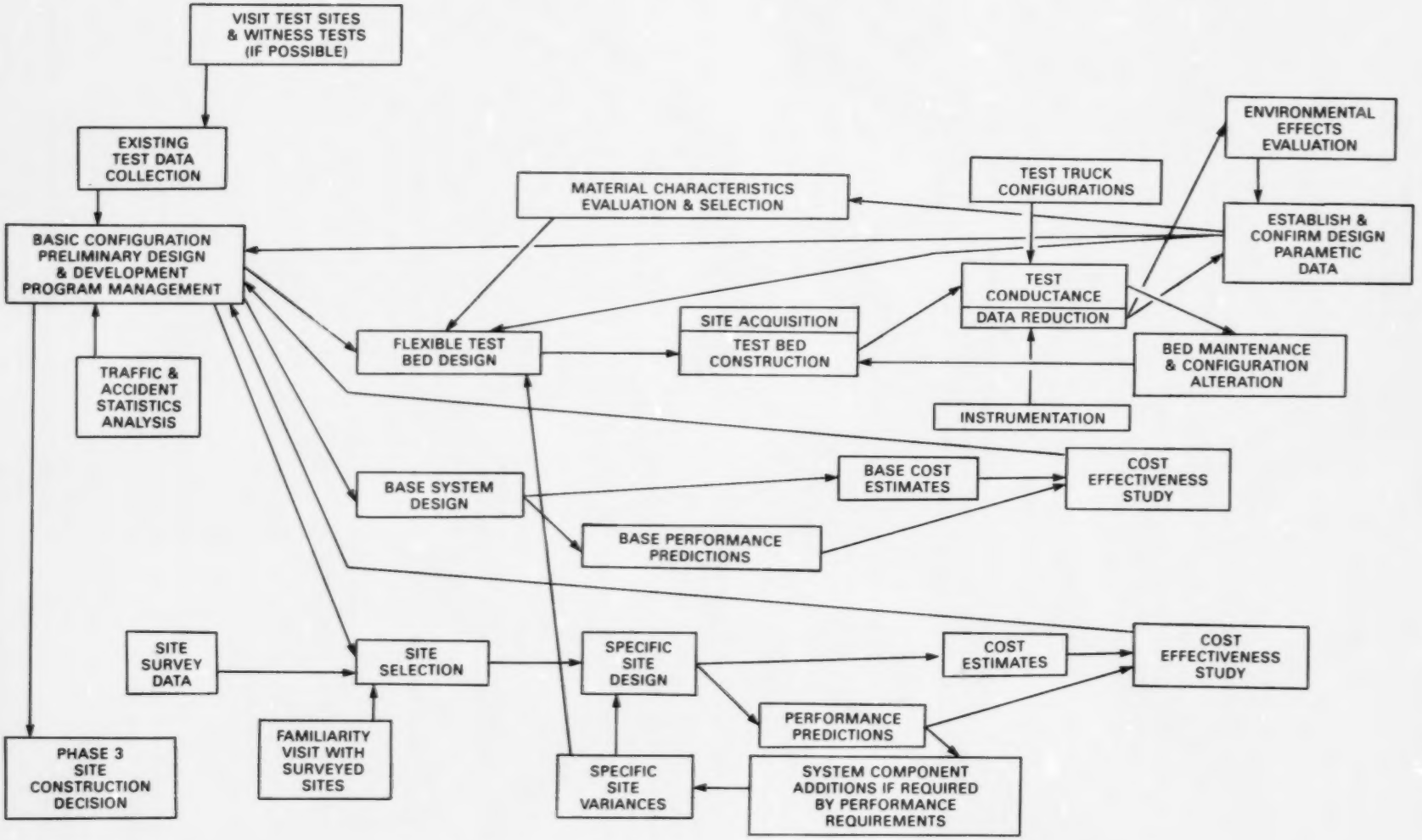
- **Passive System**
  A passive system, for example a gravel bed arrester, is the most attractive of the arresters studied. It is basically simple, enjoys the lowest of the system installation and maintenance costs and requires little upkeep. The gravel bed system is theoretically independent of the runaway vehicle weight and consequently vehicles of all weight classes can be equally stopped. It is expected that the drag coefficient will be affected to some degree by vehicle configuration and style. Nevertheless, since it will arrest all vehicles in about the same manner and is the only system studied exhibiting this nature, it is considered the most likely candidate for system development.

The performance estimates and comparisons of this exploratory development study were analytical. No test data were available for use as checkpoints. The mathematical modelling was simplistic and the numerical integration was by trapezoidal equations. Integration intervals ranged from 0.1 to 1.0 seconds. All reactions were considered instantaneous. Iterative steps were conducted at least three times for numerical closure. No factors for conservatisms were included. It is therefore felt that the results presented are optimistic.

No formal cost estimates were made but generalized ranking can be stated. As all of the systems require a lane bed with lateral constraint to prevent vehicle jackknifing and overturning, the cost of the gravel bed passive system is, at least to ROM estimates, the base cost for all of the competitive systems. To this base cost must be added the cost of the chains, pits and barrier for chain arrester. The engine brake in turn requires the cost of the braking engines and their installation to be added onto chain system costs for ROM estimates. Now, if the engine brake is considered cost effective for the purpose for which it was developed; viz, arresting fighter aircraft worth 20 to 40 millions of dollars each, then for truck arrestment, where the amortized annual costs may approach or even exceed the possible damage costs it can be argued that the system is most certainly not cost effective. Relative to this system, the passive system is by far the most cost effective. These statements notwithstanding, each critical site must be considered on its own merits before arrester siting decisions are made.

8.2 Recommendations

It is recommended that development and preliminary design proceed on the basic passive (gravel bed) arrester concept. The proposed development plan is shown in Figure 8-1.



PASSIVE ARRESTER SYSTEM PROPOSED DEVELOPMENT PLAN - PHASE 2 TECHNICAL

FIGURE 8-1

APPENDIX

Runaway Truck Arrester System Study
Technical Specifications

The study should be based on the following parameters:

1) Maximum entry speed of vehicle into arrester system to be 160 km/hr. with normal expected speeds up to 145 km/hr.
2) The deceleration should be progressive (linear or nearly linear) from the initial of zero to a maximum of 2 g at the extremity of the arrester system, when the vehicle is brought to a complete stop. The case of a deceleration from zero to 3 g should also be investigated for even shorter stopping distances, should terrain dictate this as a requirement.
3) Although prime consideration should be given to arresting such vehicles as trucks, semi-trailers or double trailers in the gross weight range of 65,000 kg to 7,000 kg, the effects upon a small vehicle in the gross weight range up to 2,000 kg should also be examined, in case one of these smaller vehicles inadvertently used the arrester system, or a heavy hauler of up to 110,000 kg.
4) The arrester system should require minimum maintenance and should be capable of being installed in remote areas where utilities may not be easily available.
5) The arrester system should be such that possibility of roll back after stopping is minimized.
6) The arrester system must be capable of operation throughout a temperature range of -40°C to 35°C.